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THE JOURNAL OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

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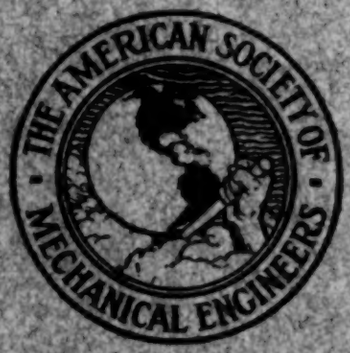

· SEPTEMBER · 1915 ·

SAN FRANCISCO MEETING, SEPTEMBER 16-17
ANNUAL MEETING, NEW YORK CITY, DECEMBER 7-10

THE OPPORTUNITY OF THE ENGINEER

THE Secretary of the Navy of the United States has invited the Society to appoint representative engineers to serve on an Advisory Board. Likewise officers of the War Department have solicited the cooperation of the Society in the organization of a Civilian Reserve Corps.

The national organizations of technically trained men are ideally equipped to render service. The Society is prepared to serve similarly many other departments of the government and is promulgating this idea wherever opportunity offers. The conception of the Society is primarily to give its service to the profession and for the public good.

"Give to the world the best that you have
and the best will come back to you."

Service by the Society as a whole means service by each individual member. Let all closely cooperate and urge every member of the engineering profession within reach to attend the meetings of the Society wherever held, whether or not he is a member.

The season is opportune to begin this great work. In October, Local Sections will resume regular meetings in fifteen of the largest industrial centers in the country.

The Secretary will be glad to send a special invitation to any distinguished engineer whose attendance is desired by any member in order to contribute to the pleasure and success of a meeting.

Total Membership of the Society, August 26, 1915 6604
New Members since January 1, 1915 413

THE JOURNAL OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

Volume 37 September 1915 Number 9

THE SAN FRANCISCO MEETING AND THE INTERNATIONAL ENGINEERING CONGRESS, 1915

AS the time approaches for the September meeting and the International Engineering Congress, interest is centered on the programs of papers to be presented and on the other activities that have been scheduled. A complete account appeared in the August issue of The Journal of the preliminary affairs in connection with the meetings, such as the transportation arrangements, the headquarters and hotel accommodations, the meetings of other societies, the excursions, the Exposition, etc., but little was definite relative to the program of the meetings at that time. The programs of both the September meeting of the Society and of the International Engineering Congress have now been established and they are presented herewith in complete form.

PROGRAM OF THE SEPTEMBER MEETING

Headquarters at Clift Hotel, San Francisco, September 16 and 17.

Meetings in Hall of the Native Sons of the Golden West, Mason Street, between Geary and Post Streets.

THURSDAY, SEPTEMBER 16.

Address of welcome by a leading citizen of San Francisco and response by the Vice-President, G. W. Dickie.

ENGINEERING FEATURES OF THE PANAMA-PACIFIC INTERNATIONAL EXPOSITION, G. L. Bayley.

MECHANICAL ENGINEERING AT THE PANAMA-PACIFIC INTERNATIONAL EXPOSITION, G. W. Dickie.

FRIDAY, SEPTEMBER 17.

THE HEAVY OIL ENGINE, ITS PRESENT STATUS AND FUTURE DEVELOPMENT, A. H. Goldingham.

THE DIESEL ENGINE AND ITS APPLICATION IN SOUTHERN CALIFORNIA, W. H. Adams.

THE STRENGTH OF GEAR TEETH, G. H. Marx and L. E. Cutter.

TRANSPORTATION

As announced in The Journal for August, arrangements have been made for a special train from New York to San Francisco, for members of the Society and their friends, leaving New York (Grand Central Terminal) on Thursday, September 9, at 7.45 P.M. The train will stop at Niagara Falls for four hours on Friday morning, to permit of sightseeing. On Sunday morning the party will arrive at Colorado Springs, where side trips may be made to Crystal Park, Pike's Peak, Garden of the Gods, Cheyenne Canyon and the Seven Falls. The Grand Canyon will be reached on Tuesday morning and fifteen hours will be spent there, giving ample time for excursions to Bright Angel Trail, Bottom of Canyon, Hopi Point, Hermit Rim Drive and Sunset Point. The party will arrive at San Francisco on Wednesday, September 15, at 9 A.M.

Reservations for this train are now being arranged by Mr. G. S. Harner, Passenger Agent, New York Central Lines, 1216 Broadway, New York City. The fare to San Francisco by this train and return by any route (except via Northwestern points) is $98.80. Pullman rates, one way, are quoted as follows: lower berth, $22.00; upper berth, $17.60.

For those members of the Society who cannot join the party from New York, reservations are being made on the Sunset Limited, leaving New Orleans on Sunday, September 12, at 11 A.M., and arriving in San Francisco at 1.00 P.M. on Wednesday, September 15. Accommodations on this train can now be secured by addressing Mr. J. H. R. Parsons, General Passenger Agent, Southern Pacific Company, New Orleans, La. The round trip fare, New Orleans to San Francisco, is $57.50 going and returning over the same route. Pullman rates, one way, are quoted as follows: lower berth, $11.50; upper berth, $9.20.

HEADQUARTERS

The headquarters of the Society will be the Clift Hotel, at the corner of Geary and Taylor Streets, San

Francisco. For members who desire to be accommodated at this hotel, a number of rooms have been arranged for at a rate of $5.50 per room, including bath, to be occupied by one or two persons. Each member should make his own reservation direct with the hotel, mentioning the Society.

For convenience, the rates at other hotels as quoted to the Society are given below:

Palace Hotel, Market and New Montgomery Streets, $4.00 per day, room and bath, one person.

Fairmont Hotel, California and Mason Streets, $4.00 per day, room and bath, one person.

Hotel St. Francis, Geary and Powell Streets, $7.00 per day up, room and bath, two persons.

Hotel St. Regis, 83 Fourth Street, $2.50 per day up, room and bath, two persons.

Information regarding accommodations at any other hotels may be obtained by addressing the Official Exposition Hotel Bureau, 702 Market Street, San Francisco. This Bureau also undertakes to make reservations in approved hotels without charge.

It is also worthy of note that the Committee on Local Affairs in San Francisco has placed its services at the disposal of all engineers resident within the United States, so that those who visit San Francisco with their families and friends may be sure that special attention will be paid to their comfort.

INTERNATIONAL ENGINEERING CONGRESS, 1915

The efforts of the Committee of Management of the Congress have been indefatigable to secure the success of the enterprise by the coöperation of members from the national engineering societies. The last report of this committee announced that practically three thousand members had been enrolled and 240 papers have been received.

Members of the Society still have an opportunity of enrolling in the Congress if they desire to do so. The fee, which is five dollars, entitles the subscriber to a certificate of membership, to participation in the deliberations of the Congress, to an index volume of the Proceedings, and to one of the volumes to be published by the Congress. Remittances may be made to W. A. Cattell, Secretary, Foxcroft Building, San Francisco.

The Congress will publish ten volumes of proceedings on the topics under which the program is divided as follows: The Panama Canal; Waterways, Irrigation; Municipal Engineering; Railways, Railway Engineering; Materials of Engineering Construction; Mechanical Engineering; Electrical Engineering, Mechanical Engineering; Mining Engineering, Metallurgy; Naval Architecture and Marine Engineering; Miscellaneous, including Military Engineering, Aeronautical Engineering, and Heating and Ventilation.

The Congress will be opened at 10 A.M. on Monday, September 20, in the new Auditorium Building, where the subsequent sessions will be held. At the opening session there will be addresses of welcome and responses, an address by General George W. Goethals, Honorary President, and the presentation of the John Fritz Medal to Dr. James Douglass. Below is presented the complete program of the Congress.

PROGRAM OF THE INTERNATIONAL ENGINEERING CONGRESS, 1915

SAN FRANCISCO, CALIFORNIA, SEPTEMBER 20-25

All sessions are to be held in the Auditorium Building, Civic Centre, Hayes and Larkin Streets, at Market Street.

OPENING GENERAL SESSION.

Monday, Sept. 20, 10 a.m.

Address of welcome by the Mayor of San Francisco.

Addresses by General Goethals, Honorary President of the Congress, by Vice Presidents and by distinguished delegates.

Presentation of John Fritz Medal to Dr. James Douglas.

THE PANAMA CANAL

GENERAL SESSION.

Monday, Sept. 20, 2 p.m.

Introductory Chapter, General G. W. Goethals.

Commercial and Trade Aspects of the Panama Canal, Emory R. Johnson.

The Working Force of the Panama Canal, Captain R. E. Wood.

Purchase of Supplies for the Panama Canal, F. C. Boggs.

Outline of Canal Zone Geology, Donald F. MacDonald.

Climatology and Hydrology of the Panama Canal, F. D. Willson.

WATERWAYS

SESSION 1.

Tuesday, Sept. 21, 10 a.m.

The Province of Waterways in the Internal Commerce and Development of a Country, Brigadier-General W. H. Bixby, U. S. A.

Artificial Waterways and Natural Channels and Bodies of Water linked by Artificial Channels, constituting Inside Routes, C. S. Riche.

The Waterway from the German Rhine through the Netherlands to the North Sea along the Rivers Rhine, Waal and Nieuwe Mass, C. A. Jolles.

Natural Waterways in the United States, Lieut.-Col. Wm. W. Harts.

SESSION 2.

Tuesday, Sept. 21, 2 p.m.

Flood Control, H. M. Chittenden.

Flood Control in China, Charles Davis Jameson.

Works for the Improvement of Navigable Estuaries, Prof. Dr. Luigi Luiggi.

The River Improvement Works in Japan, Tadao Okino.

SESSION 3.

Wednesday, Sept. 22, 10 a.m.

Dry Excavation, General G. W. Goethals.

Dredging in the Panama Canal, W. G. Comber.

Construction of Gatun Locks, Dam and Spillway, Brig. General W. L. Sibert.

Method of Construction of the Locks, Dams and Regulating Works in the Pacific Division of the Panama Canal, S. B. Williamson.

SESSION 4.

Wednesday, Sept. 22, 2 p.m.

Lock Gates, Chain Fenders and Lock Entrance Caissons, Henry Goldmark.

Emergency Dams above Locks of the Panama Canal, T. B. Monniche.

Design of the Lock Walls and Valves of the Panama Canal, L. D. Cornish.

SESSION 5.

Thursday, Sept. 23, 10 a.m.

The General Design of the Locks, Dams and Regulating Works of the Panama Canal, Brig. General H. F. Hodges.

The Design of the Spillways of the Panama Canal, E. C. Sherman.

Hydraulics of the Locking Operations of the Panama Canal, R. H. Whitehead.

IRRIGATION

SESSION 6.

Thursday, Sept. 23, 2 p.m.

Irrigation Enterprise in the United States, C. E. Grunsky.

Economic Advisability of Irrigation, F. H. Newell.

Distribution Systems, Methods and Appliances in Irrigation, J. S. Dennis, H. B. Muckleston and R. S. Stockton.

Italian Irrigation, Prof. Luigi Luiggi.

Irrigation in Lybia (Italian Colony), Prof. Luigi Luiggi.

SESSION 7.

Friday, Sept. 24, 10 a.m.

Utilization of Underground Waters, G. E. P. Smith.

Irrigation in India, M. Nethersole.

Dams, Arthur P. Davis, D. C. Henny.

Earthen Dams, William Lumisden Strange.

SESSION 8.

Friday, Sept. 24, 2 p.m.

The Distribution of Water in Irrigation in Australia. Elwood Mead.

The Co-relation between Annual Demand and Supply from Natural Flow, with a Study of the Amount of Storage Necessary, L. C. Hill.

Irrigation in Spain,Distribution Systems, Methods and Appliances, J. C. Stevens.

Irrigation in Spain, Regulations Controlling the Use of Water, Metering Water for Irrigation and Methods of Charging, J. C. Stevens.

SESSION 9.

Saturday, Sept. 25, 10 a.m.

The Problem of Irrigation in the Argentine Republic, Carlos Wauters.

Duty of Water in Irrigation, Samuel Fortier.

Drainage as a Correlative of Irrigation, C. G. Elliott.

MUNICIPAL ENGINEERING

SESSION 1.

Tuesday, Sept. 21, 2 p.m.

City Planning, Nelson P. Lewis.

London Traffic in 1913, Sir Albert Stanley.

Transit Problem in American Cities, W. F. Reeves.

SESSION 2.

Wednesday, Sept. 22, 10 a.m.

Recent Progress and Tendencies in Municipal Water Supply in the U. S., J. W. Alford.

Municipal Water Supply in France, Belgium, Algeria-Tunisia, Dr. E. Imbeaux.

The Disposal of Suspended Matters in Sewage, Rudolph Hering.

Sewage for Low Countries with Special Regard to the Town of Amsterdam, A. W. Bos.

SESSION 3.

Wednesday, Sept. 22, 2 p.m.

Streets, George W. Tillson.

Rural Highways, L. W. Page.

Rural Highways, L. Limasset.

Construction and Maintenance of Rural Highways, Alfred Dryland.

Rural Highways, Arthur Gladwell.

The Struggle Against Dust, C. C. Dassen.

SESSION 4.

Friday, Sept. 24, 10 a.m.

Utilities, Dr. A. C. Humphreys.

Short Paper on Public Utilities, Edward Willis.

Arch Bridges of Hooped Cast Iron, Dr. Ing. Fritz von Emperger.

SESSION 5.

Friday, Sept. 24, 2 p.m.

Preliminary Municipal Engineering at Panama, Henry Welles Durham.

Municipal Engineering and Domestic Water Supply in the Canal Zone, George M. Wells.

Sanitation in the Panama Canal Zone, Charles F. Mason.

"Soliditit" Concrete Roads in Italy, Prof. Luigi Luiggi.

RAILWAY ENGINEERING

SESSION 1.

Tuesday, Sept. 21, 10 a.m.

The Status of the Railways of North and South America, F. Lavis.

Italian Railways, Prof. Luigi Luiggi.

The Status of Indian Railways, Victor Bayley.

The Status of Chinese Railways, Charles Davis Jameson.

The Status of Russian Railways, V. A. Nagrodsky.

The Status of Railways and Tramways in the Netherland East Indies, E. P. Wellenstein.

SESSION 2.

Tuesday, Sept. 21, 2 p.m.

Economic Considerations Controlling and Governing the Building of New Lines, John F. Stevens.

The Locating of a New Line, William Hood.

The Reconstruction of the Panama Railroad, Frederick Mears.

The Locating of a New Line, David Wilson.

Railway Construction Methods and Equipment in Australia, Maurice E. Kernot.

SESSION 3.

Thursday, Sept. 23, 10 a.m.

American Railroad Bridges, J. E. Greiner.

Tunnels, Charles S. Churchill.

Tunnels in Italy, Prof. Luigi Luiggi.

Tunnels in Switzerland, R. Winkler.

Track and Roadbed, George H. Pegram.

SESSION 4.

Thursday, Sept. 23, 2 p.m.

Recent Locomotive Development, George R. Henderson.
Rolling Stock (other than Motive Power), A. Stucki.
The Floating Equipment of a Railroad, F. L. DuBosque.
Railroad Terminals, B. F. Cresson, Jr.

SESSION 5.

Friday, Sept. 24, 10 a.m.

Electric Motive Power in the Operation of Railroads, William Hood.
Electric Motive Power in the Operation of Railroads, E. H. McHenry.
Signals and Interlocking, Charles Hansel.

MATERIALS OF ENGINEERING CONSTRUCTION

SESSION 1.

Tuesday, Sept. 21, 10 a.m.

Structural Timber in the United States, H. S. Betts, W. B. Greeley.
Timber in Canada, R. H. Campbell.
Indian Timbers used in Engineering Construction, R. S. Pearson.
Timber in Russia, Mr. Tkachenko.
Preservative Treatment of Timber, Howard F. Weiss, Clyde H. Teesdale.

SESSION 2.

Tuesday, Sept. 21, 2 p.m.

Clay Products as an Engineering Material, A. V. Bleininger.
Aggregates for Concrete, S. E. Thompson.
Probable and Presumptive Life of Concrete Structure made from Modern Cements, Bertram Blount.
Volume Changes in Concrete, Alfred H. White.
Use of Wood and Concrete in Structures standing in Sea Water, with Special Reference to Dock Work, Harrison S. Taft.

SESSION 3.

Wednesday, Sept. 22, 10 a.m.

The Outlook for Iron, Prof. James Furman Kemp.
Alloy Steels in Bridgework, J. A. L. Waddell.

SESSION 4.

Thursday, Sept. 23, 10 a.m.

The Economics of the World's Supply of Copper, Thomas T. Read.
Consumption of Copper and its various Uses, Thomas T. Read, H. D. Hawks.
Alloys and their Use in Engineering Construction, W. Reuben Webster.
The Engineering Uses of Aluminum, Prof. Jos. W. Richards.

SESSION 5.

Thursday, Sept. 23, 2 p.m.

Testing of Materials, R. G. Batson.
Testing Full Size Members, Gaetano Lanza.
Proof Testing of Structures, J. E. Howard.

MECHANICAL ENGINEERING

SESSION 1.

Tuesday, Sept. 21, 10 a.m.

Recent Advances and Improvements in Founding, Thomas D. West (deceased).
Forgings from the Early Times until the Present, C. von Philip.
Recent Progress and Present Status of the Art of Forging with Special Reference to the Use of Quick-Acting Forging Presses, A. J. Capron.
Permanent Shops, Pacific Terminals, Panama Canal, H. D. Hinman, A. L. Bell.

SESSION 2.

Tuesday, Sept. 21, 2 p.m.

Machine Shop Equipment, Methods and Processes, E. R. Norris.
Machine Shop Equipment, Methods and Processes, H. F. L. Orcutt.
Automatics, R. E. Flanders.
High Temperature Flames in Metal Workings, H. R. Swartley, Jr.

SESSION 3.

Wednesday, Sept. 22, 10 a.m.

The Gas Power System. A Survey of its Status in the Year 1915, Professor Charles Edward Lucke.
The Development of the Construction of Turbines in the Netherlands, D. Dresden.
The 1915 Steam Turbine, E. A. Forsberg.
The Diesel Engine in America, Max Rotter.

SESSION 4.

Thursday, Sept. 23, 2 p.m.

Developments in Modern Water Turbine Practice, Dr. H. Zoelly.
Water Wheels of Pressure Type, Arnold Pfau.
Hydraulic Power Development and Use, J. D. Galloway.
Water Wheels of Impulse Type, W. A. Doble.
Canadian Hydraulic Power Development, Charles H. Mitchell.

SESSION 5.

Friday, Sept. 24, 10 a.m.

Safety Engineering, Frederick Remsen Hutton.
Motor Vehicles, Passenger Type, Ethelbert Favary.
Motor Vehicles, Utility Type, A. J. Slade.
Motor Tractors, F. S. Davis.

SESSION 6.

Friday, Sept. 24, 2 p.m.

The Boiler of 1915, Arthur D. Pratt.
Compressed Air in the Arts and Industries, W. L. Saunders.
Equipment, Processes and Methods for Boiler Shop, E. C. Meier.

ELECTRICAL ENGINEERING

SESSION 1.

Wednesday, Sept. 22, 2 p.m.

Economics of Electric Power Station Design, H. F. Parshall.
The Water Power of Sweden, Sven Lubeck.
Electric Power in Canadian Industry, Charles H. Mitchell.
The Effect of Hydro-Electric Power Transmission upon Economic and Social Conditions, Frank G. Baum.

SESSION 2.

Thursday, Sept. 23, 10 a.m.

Electrical and Mechanical Installations of the Panama Canal, Edward Schildhauer.
Electric Welding, C. B. Auel.
The Application of Electricity to the Heating of Metals, F. L. Bishop.
The Electric Motor as an Economic Factor in Industrial Life, David B. Rushmore.

SESSION 3.

Thursday, Sept. 23, 2 p.m.

The Influence of the Electric Motor on Machine Tools, A. L. DeLæuw.
Effects of Electrolysis on Engineering Structures, Albert F. Ganz.
The Mechanical Problem of the Electric Locomotive, G. M. Eaton.
On the Production of High Permeability in Iron, Ernest Wilson.

MINING ENGINEERING

SESSION 1.

Tuesday, Sept. 21, 10 a.m.

Economic and Social Influence of Mining, W. H. Shockley.
Valuation of Metal Mines and Prospects, T. A. Rickard.
The Valuation of Oil Lands and Properties, M. E. Lombardi.
Valuation of Coal Mines and Lands, R. V. Norris.
Valuation of Anthracite Mines, R. V. Norris.
Valuation of Coal Lands, Samuel A. Taylor.
Evaluating Coal Properties in Western Canada, R. W. Coulthard.
Valuation in France, E. Gruner.

SESSION 2.

Wednesday, Sept. 22, 10 a.m.

Functions and Work of Exploration and Development Companies, H. W. Turner.
European Mining Finance, J. L. Gallard.
The Financing of Mines in the United States, Lucius W. Mayer.

SESSION 3.

Wednesday, Sept. 22, 2 p.m.

Organization and Staff of Mining Companies, W. H. Shockley, R. E. Cranston.
Relations of Governments to Mining, Horace V. Winchell.
Mine Inspection, J. W. Paul.

METALLURGY

SESSION 1.

Tuesday, Sept. 21, 2 p.m.

Symposium on Iron and Steel, edited by Henry M. Howe.
Iron and Steel Castings, John Howe Hall.
Metallography and the Hardening of Steel, Albert Sauveur.
Case Hardening of Steel, F. Giolitti.
The Duplex Process of Steel Manufacture, F. F. Lines.
Methods of Preventing Piping in Steel Ingots, Emil Gathmann.

SESSION 2.

Thursday, Sept. 23, 10 a.m.

Symposium on Copper, edited by E. P. Mathewson.
Process in Copper Metallurgy, Thomas T. Read.
Advances in Copper Smelting, Frederick Laist.
Metallurgy of Copper in Japan, R. Kondo.
Copper Metallurgy of the Southwest, Dr. James Douglas.
Reduction Works. Copper Queen Consolidated Mining Company, Douglas, Arizona, Forest Rutherford.
Advances made in the Metallurgy of Copper, Globe District, Arizona, L. O. Howard.
Improvements in Design and Construction of Modern Copper Plants, Chas. H. Repath.

SESSION 3.

Thursday, Sept. 23, 2 p.m.

Symposium on Copper (Continued).
Leaching Copper Ores, W. L. Austin.
The Metallography of Copper, William Campbell.
Boronized Cast Copper, Dr. E. Weintraub.
Electrolytic Refined Copper, A. C. Clark.
The Development of Electrolytic Copper Refining, Lawrence Addicks.
Physical Properties of Copper, Carle R. Hayward.

SESSION 4.

Friday, Sept. 24, 10 a.m.

Symposium on Gold and Silver, edited by C. W. Merrill.
Coarse Crushing Plant; 1,000 tons capacity, G. O. Bradley.
Crushing and Grinding, L. D. Mills, M. H. Kuryla.
Solution of Gold and Silver including Classification, M. H. Kuryla.
Filtration or Separation of Metal Bearing Solution, L. D. Mills.
Precipitation, G. H. Clevenger.

SESSION 5.

Friday, Sept. 24, 2 p.m.

Symposium on the Metallurgy of Zinc, edited by Walter Renton Ingalls.
Some Main Points in the Economics of the Metallurgy of Zinc, Walter Renton Ingalls.
The Development of Zinc Smelting in the United States, George C. Stone.
The Smelting and Refining of Lead, Dr. H. O. Hofman.
Symposium on the Utilization of Fuels in Metallurgy, edited by C. H. Fulton.
Pulverized Coal in Reverberatory Furnaces, D. H. Browne.
Burning Pulverized Coal in Copper Reverberatories, E. P. Mathewson.
Gas Producer Development, Z. C. Kline.
Surface Combustion (What is it?), C. E. Lucke.
Ore Dressing, Robert H. Richards.

NAVAL ARCHITECTURE AND MARINE ENGINEERING

SESSION 1.

Tuesday, Sept. 21, 10 a.m.

Ship Calculation, Resistance and Propulsion, D. W. Taylor.
Ocean Freighters, Ernest H. Rigg.
Recent Developments in Japanese Shipbuilding, Dr. S. Terano.
Bulk Freight Vessels of the Great Lakes, Prof. Herbert C. Sadler.

SESSION 2.

Wednesday, Sept. 22, 10 a.m.

River, Lake, Bay and Sound Steamers of the United States, Andrew Fletcher.
Special Types of Cargo Steamers for the United Coast to Coast Trade through the Panama Canal, George W. Dickie.
The Development of the Sail Yacht, Steam Yacht and Motor Yacht in American Waters, William Gardner.
The Lightship, G. C. Cook.

SESSION 3.

Wednesday, Sept. 22, 2 p.m.

Warships of the First Line of Battle, Colonel E. Ferretti.
The Submarine, R. H. M. Robinson.
Present Condition of the Submarine, Max A. Laubeuf.
Modern Marine Gun Armament, H. F. Leary.
General Problem of Naval Warfare, D. W. Knox.

SESSION 4.

Thursday, Sept. 23, 10 a.m.

Marine Boilers and Boiler Room Equipment, Charles F. Bailey.
The Development of the Marine Steam Turbine, H. C. Dinger.

The Application of the Steam Turbine to Marine Propulsion, J. F. Metten.
Recent Developments in Marine Engineering in Japan, Dr. M. Tsutsumi.

SESSION 5.

Thursday, Sept. 23, 10 a.m.

Coaling Plants and Floating Cranes of the Panama Canal, F. H. Cooke.
Cargo Handling Methods and Appliances, H. McL. Harding.
Some Economic Fundamentals of Freight Handling, David B. Rushmore.
The Modern Trend in American Marine Terminals, Robt. H. Rogers.
Cargo Handling Methods and Appliances, James A. Jackson.

SESSION 6.

Friday, Sept. 24, 10 a.m.

Fuel Oil, E. H. Peabody.
The Application of Diesel or Heavy Oil Engines to Marine Propulsion, G. C. Davison.
The Diesel Motor Applied to Marine Purposes, C. Kloos.

SESSION 7.

Friday, Sept. 24, 2 p.m.

Terminal Works, Dry Docks and Wharves of the Panama Canal, H. H. Rousseau.
Aids to Navigation of the Panama Canal, W. F. Beyer.
American Graving Dock Practice, Leonard M. Cox.
Dry Docks recently built in Italy, Prof. Luigi Luiggi.

MISCELLANEOUS

SESSION 1 (Aviation).

Wednesday, Sept. 22, 10 a.m.

Aeronautics, Dr. A. F. Zahm.
Arrival of the Aeroplane, A. E. Berriman.
A Discussion concerning the Theory of Sustentation and Expenditure of Power in Flight, F. W. Lanchester.
Principles and Theories of Aerodynamics, J. C. Hunsaker.

SESSION 2 (Refrigeration).

Thursday, Sept. 23, 10 a.m.

Refrigeration, J. F. Nickerson.
Refrigeration, Thor Andersson.
Refrigeration in France, L. Marchis.

SESSION 3 (Agricultural Engrg. and Engineering Education).

Friday, Sept. 24, 10 a.m.

Agriculture and the Engineer, J. B. Davidson.
Some Observations on the Extent and Value of Farm Power Equipment, Philip S. Rose.
Some Considerations Regarding Engineering Education in America, George F. Swain.
Technical Education for the Professions of Applied Science, Ira N. Hollis.

SESSION 4 (Heating and Ventilation).

Friday, Sept. 24, 2 p.m.

Introductory Paper, R. C. Carpenter.
Recent Developments in Heating and Ventilating Art, D. D. Kimball.
Vacuum, Vacuo-Vapor and Atmospheric Heating Systems, Jas. D. Hoffman.
Recent Developments in Ventilating Machinery, W. H. Carrier.

CLOSING GENERAL SESSION.

Saturday, Sept. 25, 11.30 a.m.

MEETINGS OF OTHER SOCIETIES

The American Society of Civil Engineers, the American Institute of Mining Engineers, and the American Institute of Electrical Engineers will also hold September meetings in San Francisco in the week preceding the International Engineering Congress. The headquarters of these societies and their places of meeting are listed below:

Society	*Headquarters*	*Meetings*	*Dates*
A. S. C. E.	St. Francis Hotel	at headquarters	Sept. 16-18
A. I. M. E.	Hotel Bellevue	at headquarters	Sept. 16-18
A. I. E. E.	St. Francis Hotel	Civil Center Auditorium	Sept. 16-17

The meeting of the American Society of Civil Engineers is the Forty-seventh Annual Convention of the Society. The program includes a welcoming address by President Charles D. Marx; a Panama Canal session; a reception, dinner and dance in the Old Faithful Inn on the grounds of the Panama-Pacific International Exposition, and excursions to Del Monte, Santa Cruz and San José. The Society, through its Secretary, Chas. Warren Hunt, has extended a cordial invitation to the membership of our Society to participate in these events.

THE ENGINEERS' SPECIAL

In the July issue of The Journal reference was made to the "getting-together" arrangements for the engineers of the five national engineering societies that are to journey to San Francisco via the Engineers' special train on the New York Central, leaving New York 7.45 p.m., Thursday, September 9. A large party is scheduled for this train, the names of a considerable number of whom were listed in the July issue. Since that issue, several additional reservations have been made, and those among the membership who were not referred to in the previous list, are named below:

J. W. Upp
Henry Hess and Mrs. Hess
H. H. Barnes, Jr.
Gano Dunn
Emmett B. Carter and one
J. A. Seymour, Mrs. Seymour and Miss Seymour
William N. Best
Kenneth Seaver
E. L. Folsom and Mrs. Folsom
L. C. Marburg
Miss Kate Gleason.
H. M. Wilson and Mrs. Wilson
Charles Whiting Baker
S. Haar and party
Ralph E. Flanders
H. R. Cobleigh

PANAMA-CALIFORNIA EXPOSITION

The Exposition now being held at San Diego, California, is within easy reach of those visiting the Pacific Coast. This Exposition is also in commemoration of

the opening of the Panama Canal, but its theme is the exploitation of the possibilities and opportunities of the various sections of this coast, from Alaska to Peru.

The Exposition is staged in Balboa Park, in the heart of the city of San Diego, and its exhibits are grouped in an educational and attractive manner in twelve buildings.

PANAMA-PACIFIC INTERNATIONAL EXPOSITION

The interest of the visiting members will probably center in the Exposition and its many features of attraction. The points of particular interest to the members of the Society will be the Manufactures and Varied Interests Building; the Machinery Building; the Mines and Metallurgy Building, and the Education Building.

The great palace of Machinery is the largest on the Exposition site and is nearly a thousand feet in length. In this palace numerous groups comprising examples of steam generators and motors, internal combustion motors, hydraulic motors, and wood and metal working tools are shown.

The exhibits in the palace of Mines and Metallurgy deal with the natural mineral resources of the world, their conversion into metal and manufacture in raw materials and forms for the various industries.

Education is represented by examples including methods of vocational training and municipal training, in addition to general educational work of schools and universities.

The Manufactures and Varied Industries palace contains products of manufacture and manual skill from all nations of the world.

EXCURSIONS

For the benefit of members visiting San Francisco for the September meeting and the Congress, the following excursions to important engineering activities in California have been arranged. Some of these will be made by automobile without expense, and others will be at the cost of a round trip ticket.

September 18: San Francisco High Pressure Fire System.
Portrero Gas Works Electric Station A, Pacific Gas & Electric Co.
Spring Valley Water Works.

September 19: Delta Lands of the Sacramento and San Joaquin Rivers.
Spring Valley Water Works.

September 17 to 18: Great Western Power Company's Hydroelectric development on the Feather River.
Gold dredging at Oroville.

September 18 to 19: Pacific Gas & Electric Co.'s Hydroelectric development at Lake Spaulding and Drum Power House.
North Star & Empire Mines at Grass Valley.

September 17 to 19: Oil fields at Coalinga.

THE RETURN TRIP

Members of the party that go to San Francisco by the Engineers' Special have the privilege of returning by any one of several routes, which are given below. Rates for any of these routes will be gladly quoted by Mr. Harner, the New York Central Lines agent.

Via Southern Pacific or Western Pacific to Ogden and Salt Lake City, over the Rocky Mountains to Denver or Cheyenne, thence via Chicago or St. Louis.

Via Los Angeles, San Pedro Route to Salt Lake City, and thence as above.

Via Los Angeles, the Santa Fé Route through Albuquerque (also to Denver if desired) to Chicago.

Via Los Angeles, El Paso and the Rock Island Route to Chicago.

Via Los Angeles and El Paso to New Orleans, thence via St. Louis, Chicago or Cincinnati.

Via Portland, Tacoma, or Seattle, thence via Ogden and Salt Lake City and the Rocky Mountains to Cheyenne or Denver, and Chicago or St. Louis.

Via Portland, Tacoma and Seattle to St. Paul, thence via Chicago.

Via Portland and Tacoma to Seattle, steamer or rail to Vancouver, thence via the Canadian Rockies, Winnipeg and St. Paul to Chicago.

VOLUME 36 OF TRANSACTIONS

The annual volume of Transactions for 1914 has been distributed to the membership during the past month. This volume, like the previous ones, contains the papers and discussion given at the Annual and Spring meetings, in this case the Spring meeting at St. Paul-Minneapolis, together with reports of committees.

At the St. Paul-Minneapolis meeting there was given besides the miscellaneous papers, the symposium on powdered fuel, which is included in the volume; and among the miscellaneous papers were three contributed by local engineers on coal handling methods on the Great Lakes, flour milling and the hydraulic power development on the Mississippi River.

At the annual meeting there was a symposium on Locomotives of To-day, covering features of design and the results of operation of modern locomotives. There was also an all day meeting under the direction of the Public Relations Committee at which nine papers were presented on the general topic of the engineer in public service.

It is noteworthy that of the 45 papers and reports included in the volume, no less than 23 were prepared through the efforts of committees of experts in their respective fields, by which the authoritative character of the contributions is appreciably enhanced. There are 145 individual contributors, besides those who collaborated in the preparation of the several reports.

It was possible to include in this number of Transactions the report of the Boiler Code Committee in its

revised form, an early draft of which was sent to each member last November. This report is the most extensive, and will probably be the most far-reaching in its results, of any single undertaking by the Society up to the present time. Four states and one city have already adopted its provisions.

The variety of the papers presented and the wide scope of their contents is indicated by the following classified list of topics which were covered by the papers and discussion at the Annual and Spring meetings for 1914 and which are included in this volume:

SUBJECT CLASSIFICATIONS

Abrasives.
Air compressor unloaders.
Appraisals.
Ash fusion temperatures.
Biography.
Bonus payments.
Cast iron pipes.
Cement kilns.
Clinkering of coal.
Coal classification.
Crane wheels.
Damage cases.
Dams.
Economic problems.
Efficiency.
Electric drives.
Electric lighting plants.
Engineering training.
Ethics.
Factory buildings.
Feed water heaters.
Filters.
Fireproof construction.
Flanged fittings.
Floors.
Flour milling.
Friction losses.
Fuel.
Furnaces.
Fusion temperatures.
Gear testing machines.
Grinding wheels.
Hardening steel.
Heating boilers.
Heating value of fuels.
Hooke's joint.
Hydroelectric development.
Industrial service work.
Ingots.
Locomotives.
Materials of construction.
Meetings.
Methane.
Milling machines.
Municipal engineering.
Natural gas.
Necrology.
Police arm.
Powdered fuel.
Power plant auxiliaries.
Publicity.
Railroad equipment.
Railroad scales.
Recorders.
Reinforced concrete.
Sewage disposal.
Snow removal.
Society affairs.
Specifications.
Standardization.
Steam boilers.
Steel rails.
Stokers.
Storage of fuel.
Superheaters.
Symbols.
Testing meters.
Time studies.
Tool steel.
Transportation of fuel.
Universal joint.
Water meters.
Water power.
Weighing scales.

THE USE OF MEMBERSHIP CARDS

The membership is warned to be on guard against persons representing themselves as members of the Society. Complaints continue to be received that men posing as members are attempting to borrow money.

Membership requirements were never so strict as they are to-day, and you will protect the Society as well as yourself by requiring a stranger announcing himself a member to show the introductory card. Every member of the Society in good standing receives a card of introduction every year with the date clearly shown.

APPLICATIONS FOR MEMBERSHIP

TO BE VOTED FOR ON OCTOBER 10, 1915

Members are requested to scrutinize with the utmost care the following list of candidates who have filed applications for membership in the Society. These are sub-divided according to the grades for which their ages would qualify them and not with regard to professional qualifications, i.e., the ages of those under the first heading would place them under either Member, Associate or Associate-Member, those in the next class under Associate-Member or Junior, while those in the third class are qualified for Junior grade only. Applications for change of grading are also posted.

The Membership Committee, and in turn the Council, urge the members to assume their share of the responsibility of receiving these candidates into Membership by advising the Secretary promptly of any one whose eligibility for membership is in any way questioned. All correspondence in regard to such matters is strictly confidential, and is solely for the good of the Society, which it is the duty of every member to promote. These candidates will be balloted upon by the Council unless objection is received by October 10, 1915.

NEW APPLICATIONS

FOR CONSIDERATION AS MEMBER, ASSOCIATE OR ASSOCIATE-MEMBER

ALMEIDA, MANUEL G. ROSADO, Sales Engr., De La Vergne Mch. Co., Merida, Yucatan, Mexico

BAKER, NORMAN L., Mech. Engr., By-Products Coke Corp., Chicago, Ill.

BAKER, WM. H., Asst. Genl. Supt., The Atlas Portland Cement Co., Hannibal, Mo.

BALOUGH, CHARLES, Cons. Engr., Springfield, Ohio

BARNES, WALTER E., Vice-Pres., Consolidated Engrg. Co., Boston, Mass.

BEAKEY, PATRICK J., Foreman Type Dept., Royal Typewriter Co., Inc., Hartford, Conn.

BEAUBIEN, JAMES A., Mech. Engr., Weber Subterranean Pump Co., New York, N. Y.

BERGMAN, CARL AXEL, Palmarito de Cauto, Oriente, Cuba

BLACK, ALEXANDER L., Engr. in charge of design and construction, New Orleans Office, Ford, Bacon & Davis, New Orleans, La.

BOYD, JAMES E., Prof. of Mechanics, Ohio State Univ., Columbus, Ohio

BOYLE, WM. GEO., Mech. Supt. & Engr., Estate of Henry W. Oliver, Pittsburgh, Pa.

BREWSTER, HENRY B., Engr., with H. S. Kerbaugh, Inc., New York, N. Y.

BRUNNER, GEO. L., Treas. & Genl. Mgr., Brunner Mfg. Co., Utica, N. Y.

BRYAN, ARTIS CHESTER, Chief Engr. & Factory Mgr., Lefever Arms Co., Syracuse, N. Y.

CHACE, ERNEST M., Wks. Mgr., The Cincinnati Milling Mch. Co., Cincinnati, Ohio

CHARLES, LA VERN JOHN, Civil Engr., Constr. Engr., U. S. Reclamation Service, Elephant Butte, New Mexico

CLARK, HENRY D., with Alloy Steel Spring Co., Jackson, Mich.

COLLISON, THOMAS A., Mech. Engr., Mt. Vernon Car Mfg. Co., Mt. Vernon, Ill.

CUNNINGHAM, GEORGE F., Owner, Cunningham Construction Co., Blissville, Ark.

DOYLE, EDMUND M., Mgr., Doyle Estate Co., San Francisco, Cal.

ELY, HAROLD F., Mech. Draftsman, Alberger Pump & Condenser Co., New York, N. Y.

ENGLERT, ALFRED, Mech. Engr., Lidgerwood Mfg. Co., New York, N. Y.

FITTS, EDWIN, Engr., Detroit Stoker Co., Detroit, Mich.

FORNEY, J. RAYMUND, Southern Rep., Ralston Steel Car Co. of Columbus; Baltimore, Md.

GHERARDI, BANCROFT, Engr. of Plant, American Telephone & Telegraph Co., New York, N. Y.

GINORIO, FRANCISCO R., Elec. Engr., Cuban-American Sugar Co., Central Delicias, Oriente, Cuba

GRANGER, ABBOTT DEAN, Pres., A. D. Granger Co., New York, N. Y.

GRAUTIER, LESLIE V., Dist. M.P. Insptr., Pittsburgh Dist. & Chicago Div., Baltimore & Ohio Railroad, Pittsburgh, Pa.

GUIGNARD, C. G., Columbia, S. C.

HADLEY, SAMUEL L., Asst. Dir. of Sales, Fairbanks, Morse & Co., Chicago, Ill.

HALL, ROLAND B., JR., Mgr. New York Office, Harrisburg Fdry. & Mch. Co., New York, N. Y.

HENDERSON, MALCOLM MCG., Chief Mech. Engr., Australian Commonwealth Rwys., Melbourne, Australia.

HENNING, WARREN K., Sales Engr., Murphy Iron Works, Chicago, Ill.

HICKS, WM. A. B., Chief Draftsman, I. P. Morris Co., Philadelphia, Pa.

HITCHCOCK, HARRY S., Treas., Baker Iron Works, Los Angeles, Cal.

HOBART, HENRY M., Cons. Engr., General Elec. Co., Schenectady, N. Y.

HOWES, BENJAMIN A., Engineer in private practice, New York, N. Y.

HOXIE, V. W., Marine Dept., The Babcock & Wilcox Co., Pacific Coast Branch, San Francisco, Cal.

HUFF, RUSSELL, Cons. Engr., Packard Motor Car Co., Detroit, Mich.

JELLETT, STEWART A., Cons. and Constr. Engr., Philadelphia, Pa.

JOHNSON, GEORGE A., Mech. Supt., with W. H. Miner, Chicago, Ill.

JONES, HAROLD C., Asst. to Vice-Pres., Inland Steel Co., Chicago, Ill.

KLEIN, BERNARD J., Exhibit Mgr., The Bristol Co. of Waterbury, Conn., Palace of Machinery, San Francisco, Cal.

LANGILLE, HERBERT B., Asst. Prof. Mch. Design and Mech. Drawing, Univ. of Cal., Berkeley, Cal.

LARKIN, FRED V., Asst. Prof. Mech. Engrg., Lehigh Univ., So. Bethlehem, Pa.

LEHMAN, WERNER, Mech. Engr., Bucyrus Co., South Milwaukee, Wis.

LEWIS, E. J., Supt., Climax Mfg. Co., Corry, Pa.

LEWIS, GEORGE WM., Asst. Prof. Mech. Engrg. in charge of Exper. Engrg., Swarthmore College, Swarthmore, Pa.

LEWIS, WARREN B., Cons. Engr., Providence, R. I.

LICHTNER, WILLIAM O., Cons. Engr., Newton Highlands, Mass.

LOOSE, THERON L., Genl. Supt., Hendee Mfg. Co., Springfield, Mass.

LYSTER, T. L. B., Chief Engr., Hooker Electrochemical Co., Niagara Falls, N. Y.

MACLEOD, NORMAN MACCALLUM, Lab. Elec. Engr., Western Electric Co., New York, N. Y.

MAAS, ELOV F., Chief Draftsman Machinery Div., Navy Yard, Puget Sound, Wash.

MANCHESTER, HENRY C., Supt. of Motive Pwr. & Equip., Delaware, Lackawanna & Western R.R., Scranton, Pa.

MATHEY, HENRY C., Asst. Engr., Underwriters' Laboratories, Inc., New York, N. Y.

METTEN, JOHN F., Chief Engr., Wm. Cramp & Sons Ship & Eng. Bldg. Co., Philadelphia, Pa.

MORSS, HENRY A., Vice-Pres., Simplex Wire & Cable Co., also The Morss & Whyte Co., Boston, Mass.

NOBLE, HOWARD A., Vice-Pres., Pittsburgh Spring & Steel Co., Pittsburgh, Pa.

PACKER, E. RAY, Mech. Engr., The Q & C Co., Chicago, Ill.

PAFFEN, PAUL J., Engr., Barrett Mfg. Co., New York, N. Y.

PARCE, JOSEPH YALE, Director Shop Work, Manual Training High School, Denver, Colo.

PROUTT, FREDERICK G., Cons. Engr., Memphis, Tenn.

RUSS, JOHN B., Mech. Supvr., The Robert N. Bassett Co., Derby, Conn.

SANDERS, W. E., Mgr. Packing Dept., Essex Rubber Co., Trenton, N. J.

SHAW, WM. M., Vice-Pres. & Genl. Mgr., The Benj. F. Shaw Co., Wilmington, Del.

SMITH, ALBERT H., Production Engr., Sefton Mfg. Co., Chicago, Ill.

SMITH, JAMES C., Supt., McNab & Harlin Mfg. Co., Paterson, N. J.

STREIFF, ABRAHAM, Hyd. Engr., Fargo Engrg. Co., Jackson, Mich.

TOMPKINS, GEORGE R., Head of Mech. Dept., Wilberforce Univ., Wilberforce, Ohio

VOCKE, CHAS. WM., Developing patent claims, Ridgewood, N. J.

WAHL, HENRY R., Mech. Engr., Commonwealth Edison Co., Chicago, Ill.

WALKER, F. W., JR., Genl. Mgr., Beaver Falls Art Tile Co., Beaver Falls, Pa.

WATKINS, EDWARD G., In charge Engrg. Dept., Heywood Bros. & Wakefield Co., Gardner, Mass.

WHITE, HAROLD E., Controller Engr., Crocker-Wheeler Co., Ampere, N. J.

WILLIAMS, ARTHUR, Genl. Commercial Mgr., The New York Edison Co., New York, N. Y.

WOODARD, WM. E., Asst. Chief Engr., Amer. Loco. Co., Schenectady, N. Y.

FOR CONSIDERATION AS ASSOCIATE-MEMBER OR JUNIOR

ABBE, ROY H., Mech. Engr., Towle Mfg. Co., Newburyport, Mass.

ADAIR, CRAIG, Mech. Engr., The Baldt Steel Co., New Castle, Del.

AIREY, JOHN, Asst. Prof. of Engrg. Mechanics, Univ. of Mich., Ann Arbor, Mich.

BRONSON, CARLOS E., Asst. Mech. Engr., Kewanee Boiler Co., Kewanee, Ill.

BUCKLEY, AMBROSE T., Electrician, Dept. of Public Charities, Staten Island, N. Y.

COBB, HOWARD P., Industrial Engr., D. B. Lewis & Co., Boston, Mass.

CURTISS, CHARLES B., with The Wickes Boiler Co., Saginaw, Mich.

DANIELS, CLARENCE W., Plants Engr., Norton Co., Worcester, Mass.

FRENCH, CHARLES M., Sales Engr., Wagner Elec. Mfg. Co., Springfield, Mass.

GRAVES, RALPH I., Mech. Asst. to Genl. Pur. Agt., Chicago & North Western Rwy. Co., Chicago, Ill.

GRUNERT, ARTHUR E., Asst. Efficiency Engr., Commonwealth Edison Co., Chicago, Ill.

HEMPEL, ALBERT G., Denver, Colo.

HEYWARD, THEODORE C., Asst. to Thomas B. Whitted, Charlotte, N. C.

NOURSE, CHESTER LINWOOD, Mech. Engr., The Fiberloid Co., Indian Orchard, Mass.

PEW, JOSEPH N., JR., Vice-Pres., Sun Co., Philadelphia, Pa.

PULLEN, ROYAL R., Mech. Engr., The Canadian Klondyke Mining Co., Ltd., Dawson, Y. T., Canada

REED, HARRY A., Resident Engr., Stone & Webster Engrg. Corp., Boston, Mass.

REEDS, CLARENCE, Cons. Engr. with John A. Stevens, Lowell, Mass.

SEES, JOSEPH FRANK, Asst. Chief, Equip. Engrg. Div., Union Metallic Cartridge Co., Bridgeport, Conn.

THOMPSON, PAUL W., Tech. Engr., The Edison Ill. Co., Detroit, Mich.

WING, STEPHEN R., Prof. Mech. & Elec. Engrg., Pennsylvania College, Gettysburg, Pa.

FOR CONSIDERATION AS JUNIOR

BLANK, BERNARD, Elec. Designer, Pennsylvania Railroad, West Jersey & Seashore Div., Camden, N. J.

BURLEY, E. R., JR., Asst. Engr., Old Reliable Motor Truck Co., Chicago, Ill.

DEDICKE, CARL E., Engr., The Felters Co., Inc., Middleville, N. Y.

DICKEY, RALPH L., Planning Clerk, Warren Power Sta., N. Y., N. H. & H. R. R., Warren, R. I.

ENDICOTT, GEORGE, with The Morgan Spring Co., Worcester, Mass.

FOLLMER, CLINTON L., Mech. Engr., The Consolidation Coal Co., Inc., Fairmont, W. Va.

FRANKLIN, PAUL A., Supt. of Newark Warehouse, American Sanitary Wks., Harrison, N. J.

GUILFORD, EDWARD F., Asst. Supt. of Dial Dept., Hamilton Watch Co., Lancaster, Pa.

HOWE, JACK L., Estimator & Salesman, Otis Elev. Co., San Francisco, Cal.

HOYT, FRANK W., Engr., Winchester Repeating Arms Co., New Haven, Conn.

LAUER, CARLETON J., Tiffin, Ohio

MACKENZIE, G. EARL, Research Dept., Western Elec. Co., New York, N. Y.

MAY, HOWELL B., Special Apprentice with American Steel Foundries, Alliance, Ohio

NEIL, EDMUND B., Personal Asst. to Chief Eng., Truck Dept., Pierce-Arrow Motor Car Co., Buffalo, N. Y.

NORRIS, J. BOYD, JR., Oiler with The Coastwise Transportation Co., Boston, Mass.

PARSELL, ROY LINWOOD, with Winchester Repeating Arms Co., New Haven, Conn.

REYNOLDS, JAMES J., Genl. Supt., Thurlow Steel Works, Inc., Chester, Pa.

SCHAEFER, WALTER A., Underwriter with Workmen's Compensation Insurance & Ocean Accident & Guar. Corp., Ltd., New York, N. Y.

SCRANNAGE, LAWRENCE E., Asst. in Planning Dept., Remington Arms & Ammunition Co., Bridgeport, Conn.

SHORT, FRANK, Instr. in Elec. Engrg., Univ. of Pennsylvania, Philadelphia, Pa.

STARK, WILLET E., Engr., The B. F. Sturtevant Co., Hyde Park, Mass.

STRADER, ROLAND HAROLD, JR. Asst. to Supt., Standard Gas Light Co., New York, N. Y.

STURMFELSZ, GEORGE J., JR., Draftsman, Maryland Steel Co., Sparrows Point, Md.

VOORHEES, ALBERT C., Head of Tech. Dept., Lake Torpedo Boat Co., West Coast Branch, Long Beach, Cal.

WEISS, HERBERT A., Asst. Prof. in Drawing, Georgia School of Tech., Atlanta, Ga.

WILTON, HUGH, Dir. of Mch. Shop Practice & Trade, State Trade School, Putnam, Conn.

APPLICATIONS FOR CHANGE OF GRADING

PROMOTION FROM ASSOCIATE

BASS, W. J., WILLARD J., Engr., Armour & Co., Chicago, Ill.

THOMPSON, O. C., Mech. Engr., Wirebounds Corp., Chicago, Ill.

PROMOTION FROM ASSOCIATE-MEMBER

MISOSTOW, HENRY, Mech. Engr., Smoke Dept., City of Chicago, Chicago, Ill.

WRIGHT, PAUL, with American Cast Iron Pipe Co., Birmingham, Ala.

PROMOTION FROM JUNIOR

BLUMENFELD, RALPH, Asst. Mech. Engr., Keller Mech. Engr. Co., Brooklyn, N. Y.

BRAMAN, SAMUEL N., with The Morss & Whyte Co., Boston, Mass.

CARTER, HAROLD T., Supt., Lucknow Water Works, Lucknow, India

WOODS, SAMUEL H., Asst. Engr., Elec. Div., General Vehicle Co., Inc., Long Island City, N. Y.

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NEW APPLICATIONS FOR LISTING IN THE 1916 YEAR BOOK

Applications should be filed by September 27, in order that those elected may qualify in time to be included in the list of members appearing in the 1916 issue of the year book.

THE CONNORS CREEK PLANT OF THE DETROIT EDISON COMPANY

BY C. F. HIRSHFELD, DETROIT, MICH.

Member of the Society

THE phenomenal growth of Detroit's population and industries has been widely heralded, but it is probable that the extent of this growth and its significance to the central station industry is not appreciated by those not closely in touch therewith. For this reason the curves in Fig. 1 are shown. The upper curve, showing the variation of population during the past decade, is probably approximately correct, because it fits smoothly into the curve obtained by plotting United States Census figures and because it checks very closely with the more accurate of the estimates which have been made from time to time. This curve indicates that the population of Detroit in 1914 was about 1.6 times as great as it was in 1904, just ten years before.

The annual output of the central stations, as given by the next lower full-line curve, has increased much more rapidly than the population. In the year 1914 it was about 21.4 times as great as in 1904. The fact that the maximum annual peak was only ten times as great in 1914 as in 1904 indicates that a very large part of the increased annual output was due to increased industrial application. This is also partly indicated by the curve showing variation of load factor.

A map of the City of Detroit is given in Fig. 2. The heavy full lines radiating, roughly, from a point near the bottom center of the figure indicate radial streets upon which the principal car lines are operated. These naturally determine the directions of growth and they indicate that, other things equal, Detroit's area may be expected to preserve a roughly semicircular shape as it expands. The heavy dotted lines indicate the rights of way of the various steam railroads entering the city. Other things equal, these rights of way, combined with the river frontage, indicate the probable future locations of the larger manufacturing industries. Apparently, these industries may be expected to scatter all along these lines in the future, as they have in the past.

The small rectangular spots indicate the locations of the various substations of the Detroit Edison Company. These are so drawn that the length of a side indicates the relative capacity. The concentration near the center from which the radial streets diverge is due to the fact that this is the business center of the city and that it is served with direct current. This gives a very concentrated and very important direct current load at this point.

The two circular spots near the lower left hand corner of the figure indicate the location of Delray power houses No. 1 and No. 2. These have an aggregate maximum capacity of about 95,000 kw. and contained practically all the generating equipment of the company before the Connors Creek plant was built. They are adjacent to one another and operated as one plant.

When it became evident about 1912 that greatly increased capacity would soon be required, two possible solutions were available; a third power house could have been built at the Delray site or a new site could have been selected. Consideration of the direction of growth of the city; the rapidly increasing population and industrial development of the east side; the location of the heavy direct current load; necessity for continuity of service with an ever increasing complexity of distribution system fed from one point near what might be called a corner of the city; and numerous similar items resulted in the choice of a new site, despite the many duplications which such a choice involved. The transmission loss in underground cables of a mean length of seven miles required to serve the growing east side manufacturing

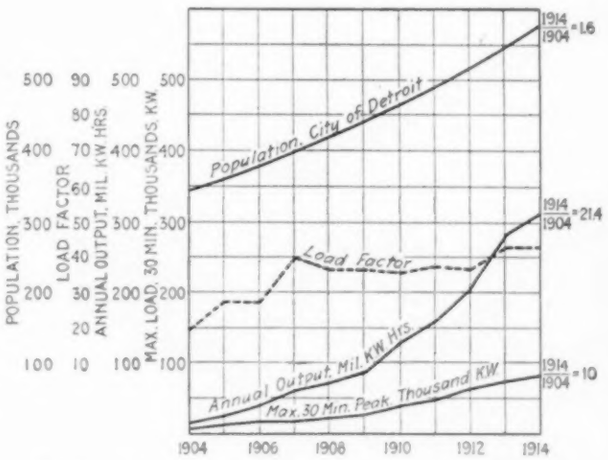

FIG. 1 CURVES INDICATING GROWTH OF CITY AND LOAD

district and the fixed charges on these cables and their conduits were, by themselves, almost adequate to determine the choice.

The site chosen is indicated by the circular dot near the lower right hand corner of Fig. 2. Being on the river front, excellent condensing water is available in enormous quantities. In this respect, it is even better situated than the older site at Delray, as the river flows from the newer toward the older site and picks up all the sewage of the city while passing the water front. Being at the river end of a terminal railroad, the new site can be supplied with coal as easily as the old site, and if water transportation of coal should become profitable, both sites are equally well located to avail thereof.

Considering the distribution problems, the new site may be said to be strategically located as a combination with the older site at Delray. A glance at the map and a study of the radiating streets and of the railroads, will show this very clearly.

The Connors Creek site is large enough to accommodate two power houses, each with greater capacity than that of the two power houses at Delray combined. The first house was planned to accommodate six 25,000 k.v.a. units, but it seems probable that this unit size will be increased before the last

Presented at the Buffalo local section of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, on March 11, 1915.

units are installed. The plan of this house is shown in Fig. 3, together with the canals and tracks which serve it. The river is located beyond the left-hand side of the figure and runs in a direction roughly parallel to the left-hand end of the plate, and upward. The diagonal position of the plant was dictated by the shape of the site, the necessity of leaving room for a future power house, the curves required for railroad tracks, and the location of the river which determined the direction and location of the canals. One-third of the first power house on the Connors Creek site has been built and two of the proposed six units are installed. The building is closed with a temporary end, so that the machinery now installed can be operated until and while future extensions are being constructed. The intake and overflow canals are completed in full size for six units.

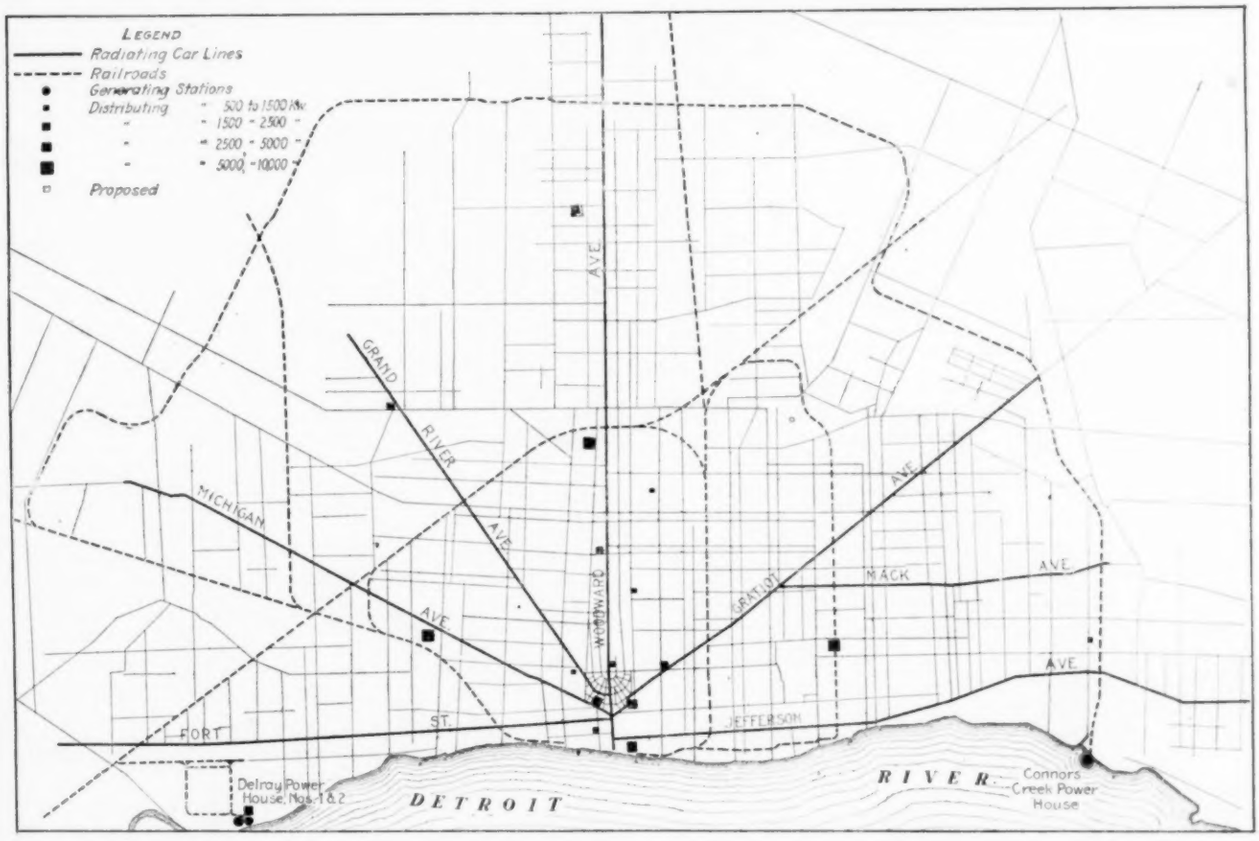

FIG. 2 MAP OF THE CITY OF DETROIT SHOWING LOCATION OF PLANTS

An architect's drawing of the finished power house is given in Fig. 4 and a sectional elevation in Fig. 5. The latter shows the construction to best advantage and should be consulted in connection with the description which follows.

The coal enters the train shed in drop-bottom cars (usually of the 50-ton size) which move on the lower group of tracks shown in Fig. 3 after being weighed on the railway scales indicated near the upper left-hand corner of that figure. The cars dump into hoppers under the tracks in the train or coal shed, there being one hopper for each unit of one turbine and two boilers. A motor-driven, variable-speed, flight conveyor with a capacity of 120 tons per hour, receives the coal from the hopper, carries it up a rather sharp incline and discharges it into a four-roll motor-driven crusher of similar capacity. This crusher breaks from 18-in. cubes, or smaller, to 100 per cent through a 1¾-in. ring.

The crusher discharges directly into a motor-driven, constant-speed, lapping bucket conveyor with 30 by 36 in. buckets. This conveyor forms an endless chain which entirely encloses the section of the boiler house, as indicated in Fig. 5. It carries the coal up on the side nearest the coal shed and discharges it into any one of the three coal bunkers which serve the two boilers of one unit. The hopper, pan conveyors, crusher and bucket conveyor for each unit of one turbine and two boilers are so located that they can deliver through chutes to one adjacent range of bunkers, serving thus as a spare for that range.

The entire coal handling equipment is electrically operated and the control is centralized at a board in the coal crushing room. Starting devices are electrically interlocked so that the various pieces of apparatus must start in proper sequence. Provision is also made to prevent starting until an observer on the floor above the coal bunkers has determined that the exposed upper horizontal run of the bucket conveyor is clear, and has so indicated. Emergency buttons are provided at numerous points along the run of the bucket conveyor; pushing any one of these stops the entire system and it cannot be started again until that particular emergency button has been reset by means of a key. The custodian of this key is therefore the only individual who can start the system after an emergency stop, and even then he must go to the point from which the stop was made.

After the coal has been dropped into the bunkers, it de-

scends by gravity to the stokers. The latter are of the underfeed, sloping fire-bed variety and there are twenty-six retorts per boiler. These are arranged in two groups of thirteen each, the boiler being fired from the two ends, as shown in Fig. 6. Ash and clinkers are removed by two clinker grinder bars located in a sort of pit running across the center of the furnace and are discharged into the ash hopper. This is an enclosure within the wind box or plenum chamber below the boiler and holds the refuse until it is finally dropped directly into standard railroad cars which run lengthwise of the house directly under the ash hoppers.

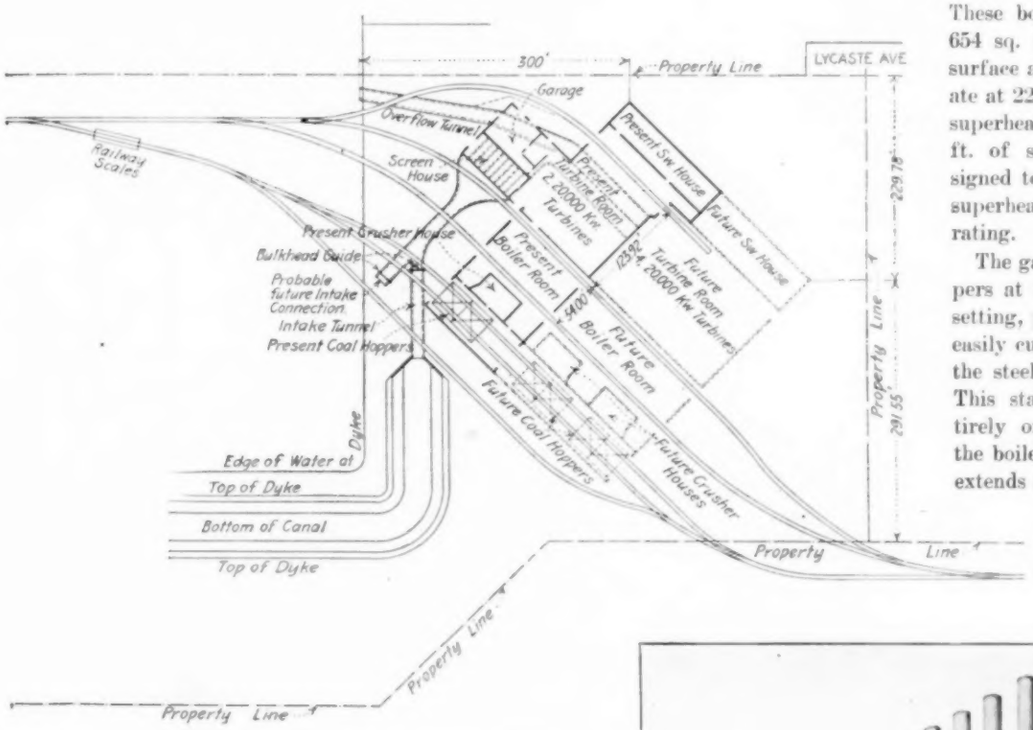

FIG. 3 PLAN OF SITE OF THE NEW POWER HOUSE

The stokers are driven through chains by direct current motors, the speed of which can be varied from 200 to 2000 r.p.m. This wide range of speed is obtained by means of armature throw over from 125 to 250 volts combined with field control. The motors are handled by means of drum controllers located at the gauge board of the particular boiler which they serve and electric tachometers are used to indicate stoker speeds at this board.

Motor driven blowers supply the air required for combustion. There are three of these blowers per range of two boilers. One of the three serves as a spare and is located between the two boilers. It is arranged to discharge either way through a Y-shaped duct. All of the blowers discharge through expanding ducts which are designed to recover the greatest possible fraction of the velocity head. These large ducts lead into a plenum chamber with a horizontal section equal to the horizontal section of the furnace. The stoker wind boxes form part of the roof of this chamber, so that the air passes directly from the plenum chamber to the stoker and does not have to pass through small ducts of any kind. Each blower is designed to deliver at maximum 74,000 cu. ft. of air per minute against a static pressure of 6.5 in. of water pressure. The motors are shunt wound and are direct connected to the blowers through flexible couplings. Their speed can be varied from 300 to 750 r.p.m. The drum type controllers are mounted at the gauge boards of the boilers served, directly under gauges which indicate the static pressure in the plenum chamber.

The boilers are similar to those used in the older plant at Delray, but are set 3 ft. higher so that the height of the combustion chamber is 33 ft. These boilers each have 23,654 sq. ft. of water heating surface and are built to operate at 225 lb. per sq. in. The superheaters have about 2400 ft. of surface and are designed to give 200 deg. fahr. superheat at 200 per cent of rating.

The gases, leaving the dampers at the top of the boiler setting, pass upward through easily curved breechings, into the steel stack near its base. This stack is supported entirely on the steel work of the boiler room structure and extends to a height of 325 ft. above the floor on which the stokers are located. It has a height of about 240 ft. above the roof of the boiler house. The stack is brick lined and has a diameter of 16 ft. inside the lining.

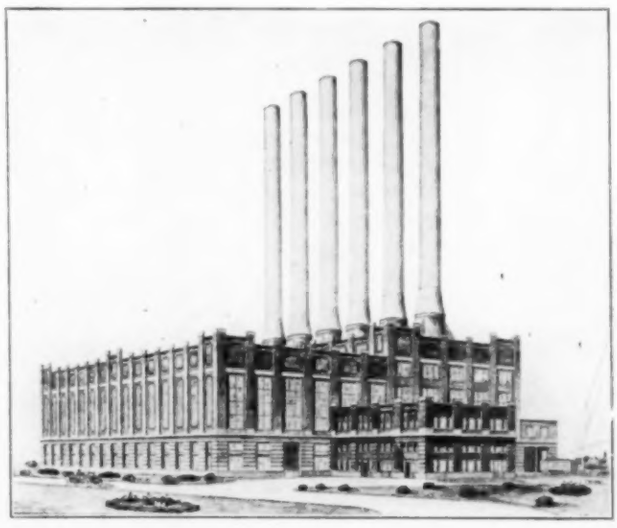

FIG. 4 VIEW OF THE PROJECTED CONNORS CREEK POWER HOUSE

As already mentioned, there are only two boilers per turbine. When operating the turbine at full load, the boilers will operate at about 170 per cent of Centennial rating. In the ordinary sense, there are no spare boilers. Experience has shown, however, that these boilers can be safely and

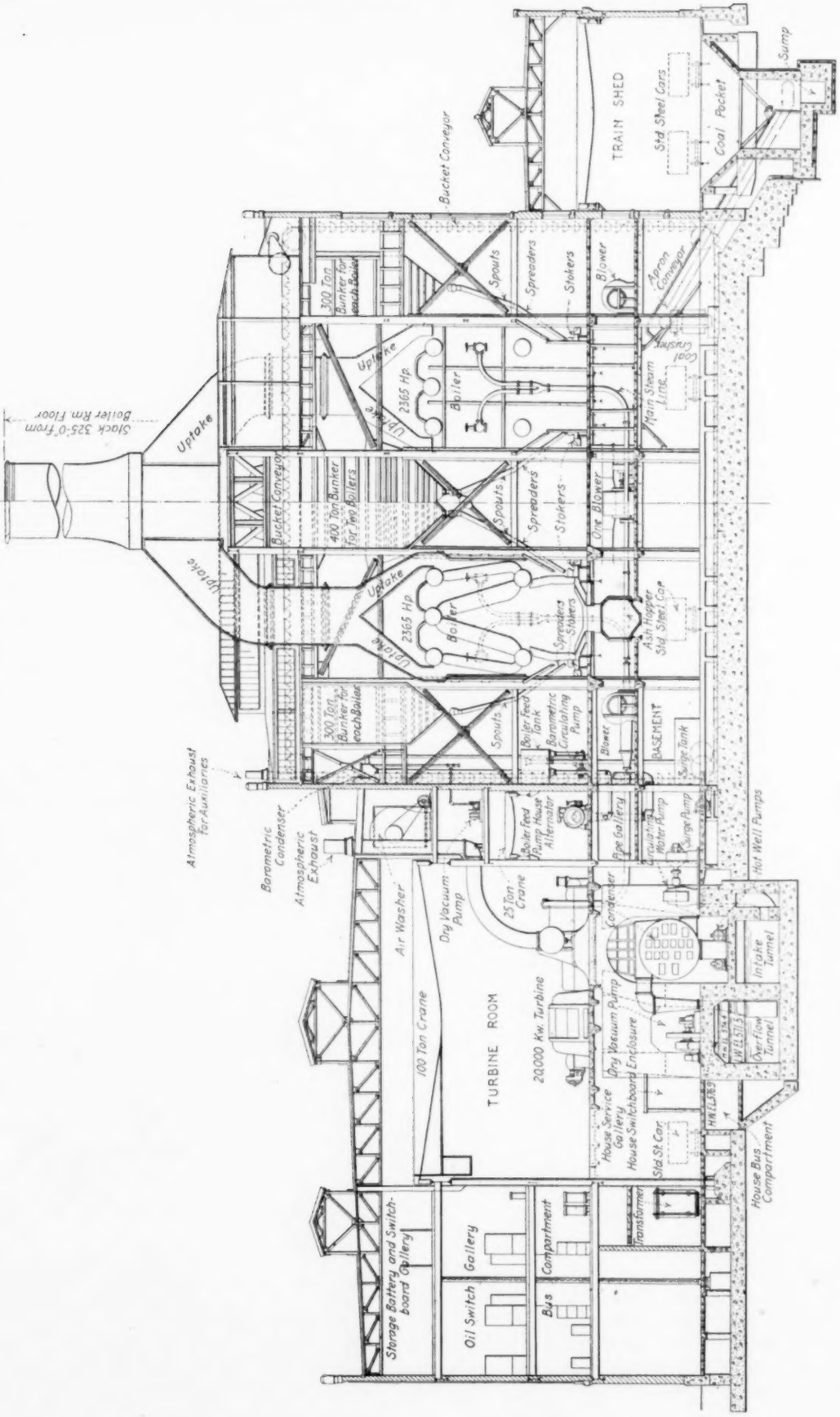

FIG. 5 CROSS-SECTION OF THE NEW CONNORS CREEK POWER HOUSE SHOWING ARRANGEMENT OF APPARATUS

economically operated at ratings considerably in excess of 170 per cent and provision has been made to enable three boilers to supply steam to two turbine units. When carrying full turbine loading under these conditions, each of the three boilers will have to operate at about 225 per cent of rating.

The main live steam piping consists of a run from two boilers to the unit which they serve, all of these runs being cross connected by a cross-over header. All of this piping is located in the so-called pipe gallery below the stoker floor, except, of course, the leads dropping down from the boilers and rising to the turbine throttles.

The steam leads from each boiler are of 10 in. pipe and these join together in a Y-fitting, which has a 14 in. discharge. Under full load conditions with two boilers supplying one unit, the steam velocities will be about 10,500 ft. per min. in the 10 in. pipe and 12,000 ft. per min. in the 14 in. pipe. With three boilers supplying two units, these velocities will rise to about 14,000 and 16,000 ft. per min., respectively.

The proposed use of the cross-over main necessitated a design which should permit steam from any two boilers to flow into that main, and steam from the main to flow into any turbine lead, with practically equal facility. After many designs had been considered, that shown in Fig. 7 was adopted. The steam leaving the 10 x 14 x 10 in. Y-branch previously mentioned, passes through a cast steel expanding nozzle which enlarges to a diameter of 28 inches. This, in turn, leads into a 28 in. cast steel side-outlet T or side-outlet cross. The 28 in. lateral outlets of the latter fittings are the connection points of the cross-over main. The velocity of the steam passing into the cross-over, or from the cross-over main to the turbine lead, is thus reduced to about one-quarter of its value in the 14 in. pipes, or roughly, a little less than 4000 ft. per min. under the worst conditions. The steam turns through the necessary right angle at this low velocity and therefore with small loss.

Steam leaving the 28 in. fitting on its way to the turbine is carried through a tapered reducer with a discharge diameter of 14 in. Similarly, steam leaving the 28 in. fitting on its way to the cross-over main passes through a tapered reducer which is cast with its longitudinal axis in the shape of a quarter circle. This reducer leads the steam into a 14 in. return bend, as shown in Fig. 7.

The steam for the auxiliary turbines, which will be mentioned later, is taken from a six inch outlet on top of the 28 in. fittings above described.

All superheated steam piping is full weight steel with welded flanges. The flanges are finished smooth and corrugated steel gaskets are used. All fittings are cast steel. At the Delray plant, Hopkinson-Ferranti valves with Venturi throats were used; these have been found to cause an excessive pressure drop because of the short length of the expanding nozzle and were therefore not deemed desirable for the new plant if an equally reliable, full opening form could be found. Full throated gate valves made by Hopkinson of the same material as the Ferranti valves were therefore used in the Connors Creek plant.

The atmospheric exhaust from the main unit is made of riveted steel pipe and fittings. The auxiliary exhaust piping is lap welded steel with Van Stone joints and fitted with corrugated copper gaskets. All valves in the exhaust lines are steel gate valves of American make. All saturated steam piping is extra heavy steel fitted with steel flanges. The fittings are all cast steel and steel valves of American make are used.

The feed water piping is extra heavy, lap welded steel, with Van Stone joints and cast steel fittings. The use of steel pipe throughout for the feed water represents a marked departure from Delray practice in which brass pipe was used for the boiler leads and connections. In laying out the feed water system at Connors Creek, the design was so ar-

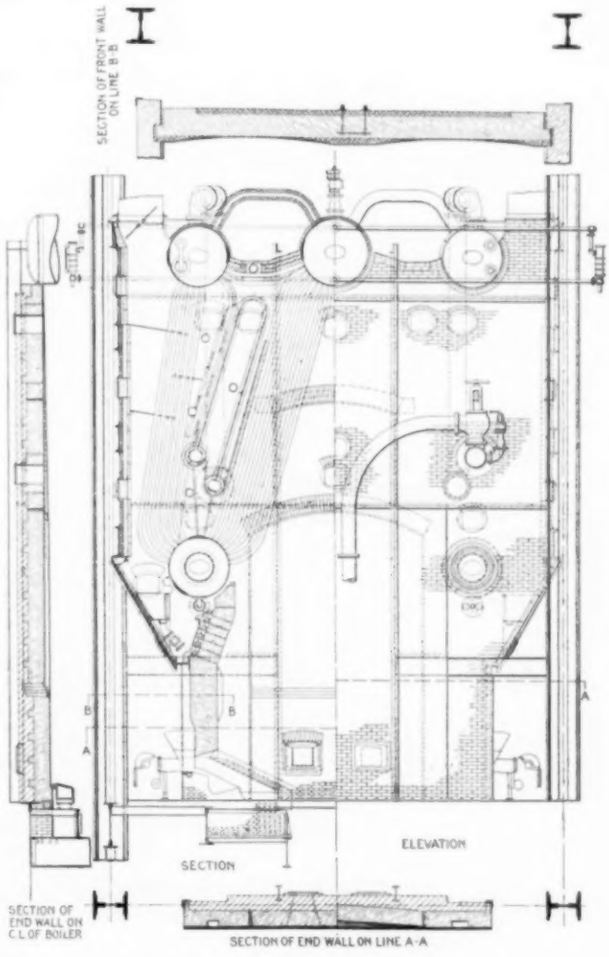

FIG. 6 DETAILS OF THE LARGE BOILERS

ranged as to permit of free expansion without the use of expansion joints of any kind.

The boiler blow-off valves are also made by Hopkinson. These were decided on after two of them had been in satisfactory operation for several years at Delray. The valve is a parallel-seat gate, on whose stem is cut a rack which meshes with a pinion. A 180 deg. turn of the handle turns this pinion through a like angle and fully opens the gate. The superiority of the valve appears to be due to the use of excellent metal and to a very high class of machine work.

The main units, as previously mentioned, are rated at 25,000 k.v.a. or 20,000 kw. at 80 per cent power factor. The steam end consists of a horizontal, nine stage, 1200 r.p.m. turbine; the electrical end generates three phase, sixty cycle current at 4800 volts. A section of the turbo-generator unit is shown in Fig. 8. The steam exhausted from the turbine

passes down through a large expanding exhaust nozzle into the condenser, as shown in Fig. 5. Each condenser is built to contain 35,000 sq. ft. of heating surface made up of 1 in. tubes a little over 18 ft. long, but has only 32,500 sq. ft. installed. The tube heads have a diameter of 14 ft. 6 in.

The tubes are so arranged as to leave numerous lanes through the steam space, so that the vapor and air readily flow to all parts of the cooling surface. No baffles of any kind are used, excepting only those necessary to form an air box. The latter is located at the bottom of the condenser, as indicated in Fig. 9.

The condensate collects in a cylindrical hot-well which extends from the lower part of the condenser shell, and it is cut shown in the plan in Fig. 3 and thence flows through a concrete tunnel to the screen house. This tunnel has a free section 9 ft. 3 in. high by 10 ft. wide, and when it is supplying water for all of the units in the finished plant under the lowest water conditions on record, the water velocity will be just under 8 ft. per second; in obtaining this, further units are assumed to be larger than those now installed.

The water flows from this section of the tunnel into a sort of funnel-shaped enlargement which leads it to the screens. These are of the traveling variety and are similar to screens already installed in the Northwest Station in Chicago and at the Delray Station in Detroit. They consist of woven-wire panels fastened to the links of an endless chain,

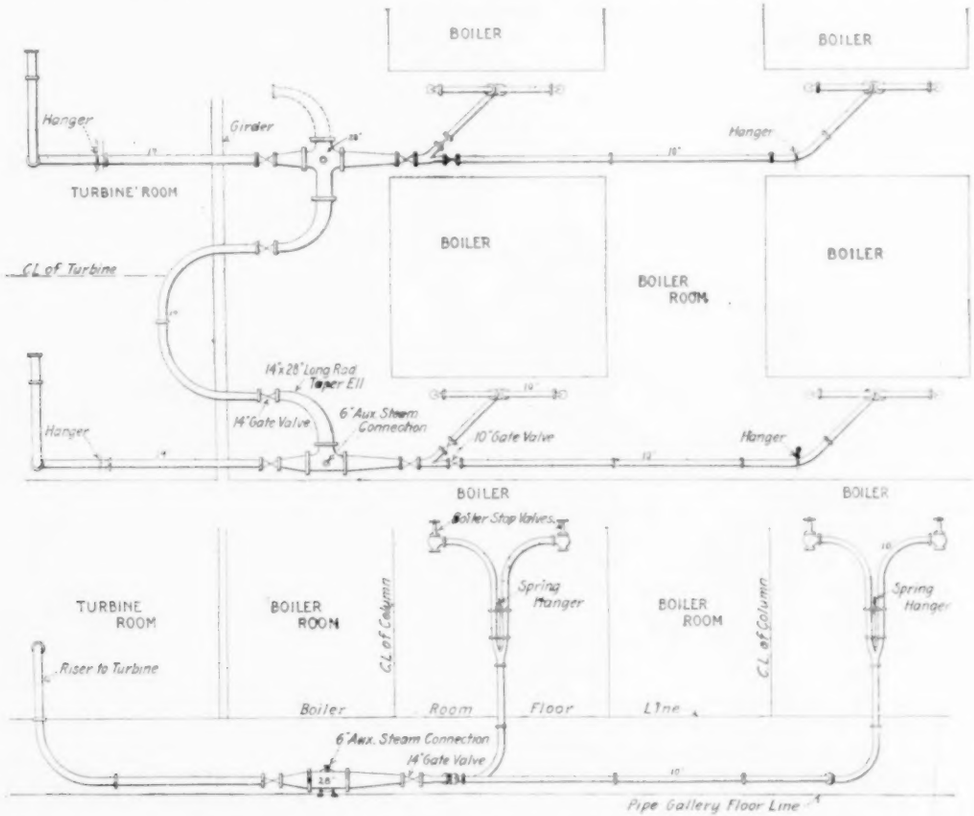

FIG. 7 DETAILS OF THE HIGH PRESSURE STEAM PIPING SYSTEM

removed by either one of two hot-well pumps. These are motor-driven two-stage centrifugal pumps operated at 1200 r.p.m. and they are fitted with bronze impellers. The condensate is discharged into what is known as the cold end of the boiler feed tank, as will be explained later.

Circulating water is supplied to each condenser by a motor-driven single-impeller double-suction, volute pump. The motor is three-phase, slip-ring type arranged for a 15 per cent speed variation, the maximum speed being 400 r.p.m. During the period of the year when the circulating water has a high temperature, the pump will be operated at the highest speed and will deliver about 40,000 gal. per min. When the water temperature is low, the pump will be operated at the lower speed and the quantity will be proportionately reduced. The temperature of circulating water varies with the season from 34 deg. to 76 deg. fahr.

The circulating water enters the property through an open

the lower end of the loop thus formed projecting down to the bottom of the water channel. The construction is shown in Fig. 10, which is a reproduction of a photograph taken in the interior of the screen house before it was completed.

When a screen becomes fouled, the working side of the loop is drawn up and over the upper guide shaft, its place being taken by what was previously the back side of the loop. This operation brings the fouled screen panels under a wash pipe which washes the fouling material into a channel arranged to carry it to the overflow canal. The screens are operated by motors and are started and stopped periodically as required. Their constant movement is only necessary under exceptional conditions. Provision is made for returning water from the overflow tunnel into the intake to assist in keeping screens and tunnel clear of ice during cold weather.

After passing through the screens, the circulating water

is turned through a right angle and flows down the length of the house in a tunnel which is 15 ft. wide and 9 ft. 3 in. high. Water is picked up by the circulating pumps through large angle fittings encased in the side wall of this tunnel.

The water passes in the condenser are rather unusual. The entering water flows through the tubes occupying one vertical half of the condenser plus those enclosed by the air box, or cooler, at the lower end of the other vertical half. The water then returns through the tubes above the air cooler and overflows through a large cast iron nozzle and elbow which points down stream in the overflow canal.

Air and non-condensible vapors are removed from a header space in the cooler by means of a 35 by 39 in. rotative, dry vacuum pump. This pump is arranged for two-stage operation in one cylinder, one end of the cylinder serving practically to load the other. A direct-current motor with armature mounted on the crank shaft drives the unit. The normal speed of the pump is 100 r.p.m., but the motor is arranged for a speed reduction to 40 r.p.m. and need seldom be operated at or near its maximum.

head of a barometric condenser, as shown in Fig. 11. The relatively cold condensate is picked up by the second pump before it has time to mix with the mass of water in the tank, and serves as injection water for the barometric condenser. The house-service alternator turbine and the boiler feed pump turbine exhaust into this barometric condenser, so that the condensate from the main unit takes up all the heat of the auxiliary steam. The foot of the barometric condenser is immersed in the other (or hot) end of the boiler feed tank. The mixture is there picked up by the boiler feed pump and delivered to the boilers.

The barometric condenser is therefore the equivalent of an open feed water heater in which exhaust steam from auxiliaries mixes with and heats the condensate from the main units, the mixture passing to the boilers as feed water. This condenser, however, offers possibilities not possessed by the ordinary open heater in that it is capable of maintaining a very low back pressure, if this is desired, or can be operated at any pressure between this and atmospheric. This property, taken in conjunction with the great variation of the

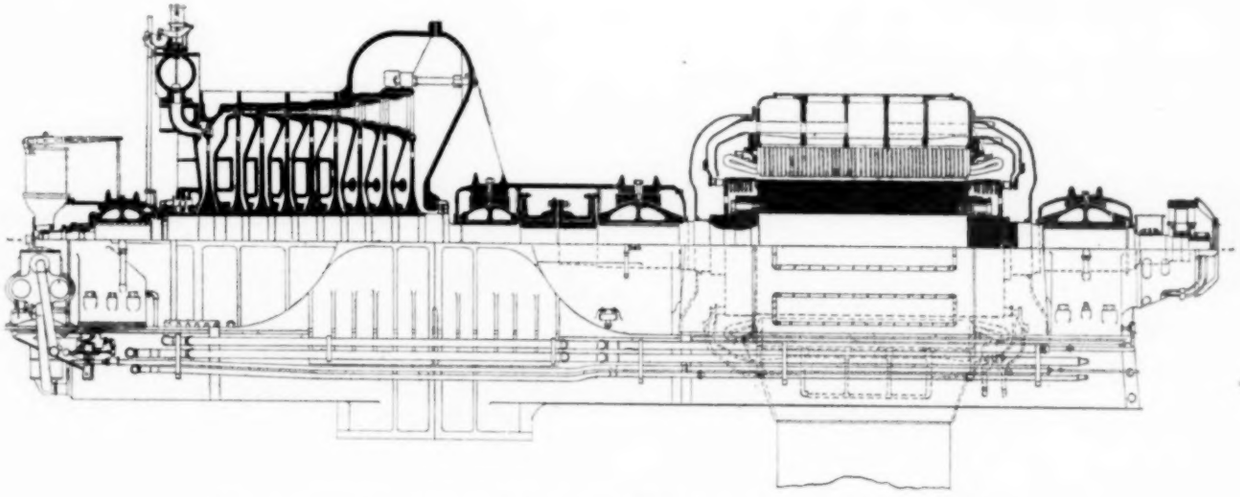

FIG. 8 SECTIONAL VIEW OF ONE OF THE 25,000 K. V. A. TURBO-GENERATOR UNITS

It will have been observed that all of the auxiliaries thus far mentioned are motor driven. As a matter of fact, this is true of all the auxiliaries in the plant with the exception of the boiler feed pumps. The latter are 1200 gal. per min., four-stage, double-suction, centrifugal pumps, driven by 350 h.p. steam turbines, at 2000 r.p.m.

The motor-driven auxiliaries may be operated, all or any of them, with power taken from the system bus, or with power taken from turbine driven, 1000 kw. alternators, known as house service alternators. In every case, that arrangement is used which, in conjunction with the other adjustments described below, will give the best heat balance for the station, under the conditions at the time existing.

What may be called the house service system is shown diagrammatically in Fig. 11. It is really a development of a method long used in both marine and stationary practice, in which a feed water heater is operated under vacuum. In the present instance, the wet vacuum pump of each main turbine unit discharges into one end of a large tank, known as the boiler feed tank. A centrifugal pump draws its water from the same end of this tank and discharges it into the

auxiliary turbine water-rate with variations of back pressure, gives great flexibility of control of the station heat balance.

This is shown diagrammatically in the upper left hand corner of Fig. 11. The straight lines represent total steam consumption of the house alternator turbines for the entire range of load at different back pressures. Assume that at some particular time with a given load and vacuum on the main unit, the load on the house alternator has the value indicated by the vertical line, and that the back pressure is that corresponding to the middle steam consumption line as at *a*. If the feed water temperature is too high it can be lowered by decreasing the back pressure, reducing the steam consumption to some such value as that shown at *c*. In this way, the feed water temperature can be accurately controlled, just enough auxiliary exhaust being made available to give the desired temperature.

The variation of the back pressure is obtained by means of a back-pressure valve in the exhaust line from the boiler feed pump turbine. If, under given constant conditions, this valve is partly closed down, the back pressure on the boiler feed pump turbine is increased and its steam consumption

for the same load is correspondingly raised. This, however, means that more steam enters the barometric condenser and, with a constant quantity and temperature of circulating water (main condensate), the temperature and pressure in the barometric condenser must rise. This, in turn, means higher back pressure on the house alternator turbine and it delivers more steam, still further raising the temperature.

Under certain conditions, this interchange may result in a cumulative temperature rise, ending only when the pressure in the barometric becomes equal to that of the atmosphere and the auxiliary exhaust blows to waste. To prevent such an occurrence, the flood valve shown in Fig. 11 is provided. This is automatically opened whenever the temperature of the steam entering the barometric condenser exceeds a predetermined value. When it operates it admits comparatively cold water from the storage tanks, adding it to the normal supply of injection water entering the condenser, thus bringing the temperature down very quickly. It then automatically shuts off. Under very poor conditions of adjustment, this valve would continue to open and shut periodically until all of the water in the storage tank and boiler feed tank had been heated to about 212 deg. fahr., but such a contingency is not probable because this would take a long time and the watch engineer would make necessary re-adjustments before the valve had acted many times.

The house alternator, boiler feed pump, and barometric injection pump are all grouped under the gallery shown at the right-hand side of the turbine room in Fig. 5. The natural position for the watch engineer is at the same place, so that continuous observation of this apparatus is readily obtained. An auxiliary gauge board containing instruments showing critical temperatures, pressures and quantities is located within easy view, and the results of all adjustments are readily seen.

This method of operating the auxiliaries appears very complicated when described on paper, but it has proved to be very simple in practice, and easy to handle. It possesses the following advantages which should be compared with methods more commonly used:

First. The one great advantage of steam driven auxiliaries is retained because exhaust steam is available for feed water heating;

Second. The quantity of auxiliary exhaust can be fitted to the ability of the feed water to absorb it;

Third. All the advantages of motor drive are obtained;

Fourth. Up to the ability of the feed water to absorb steam, the power used by the motor driven auxiliaries is produced at a thermal cost practically equal to that obtained in the case of steam driven auxiliaries;

Fifth. Since the feed water heater is normally operated under a vacuum, the feed water temperature is readily maintained at a value which will give ideal economizer operation;

Sixth. Because of the vacuum maintained in the feed water heater, air is readily removed from the feed water just before it enters the boiler feed pumps. In practice there is no reabsorption of air by the condensate between main condenser and boilers.

Another unusual feature is the provision for the distillation of all make-up water. Experience at Delray has shown that the quantity of make-up required is readily held down to 1.5 or 2 per cent. Under such conditions, the apparatus required for distillation is small and the cost is negligible in comparison with the cost of boiler labor saved. The evaporator installed has a capacity of 12,000 lb. of vapor per hour and is heated by high pressure steam. The vapor formed in it passes directly to the barometric condenser in which it mixes with the auxiliary exhaust and thus becomes part of the feed water.

Many cases of boiler pitting and corrosion which are recorded in engineering literature have been attributed to the use of very pure water and the presence of a small quantity of scale-forming material has therefore been regarded as desirable. Most authorities now seem to believe that pure water is not responsible for such damage to good boiler metal, but that the blame is to be laid on small quantities of atmospheric carbon dioxide dissolved in what is assumed to be pure water.

In the Connors Creek plant, the feed water heating and storing system has been so designed as to prevent the absorption of air, so far as prevention is readily possible. Should

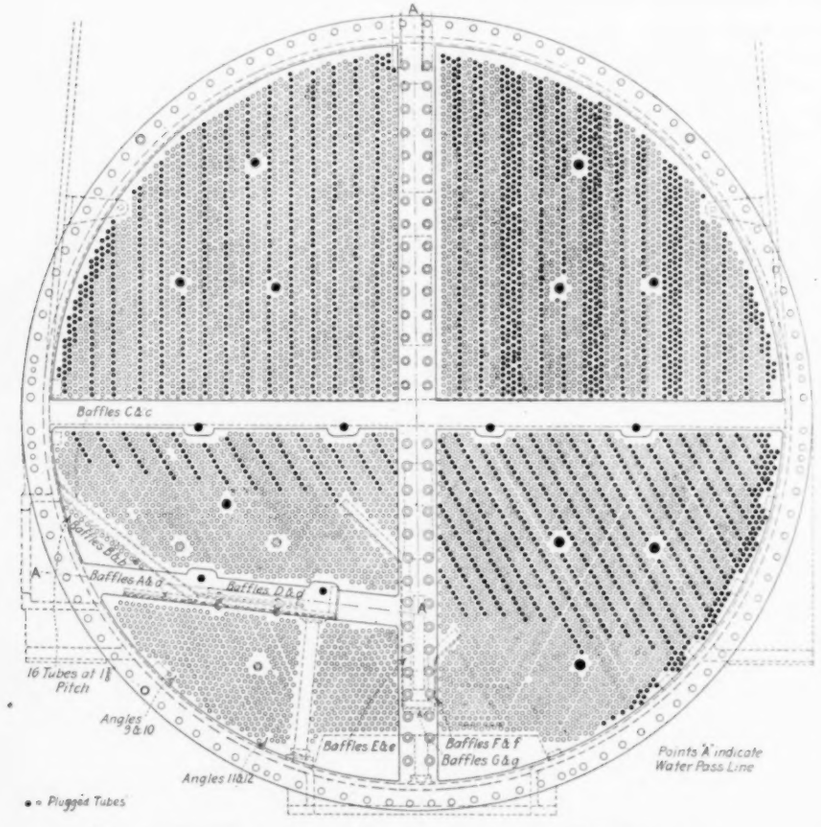

FIG. 9 DETAILS OF TUBE ARRANGEMENT IN THE CONDENSERS

experience show that the provisions made are inadequate to insure safe boiler operation, there are several simple remedies. The simplest is probably the use of a very small quantity of an alkaline salt, experience having shown that if the water shows a mere trace of alkalinity it is apparently non-corrosive.

Another innovation in this station is the very complete provision for the recovery of radiation and electrical losses. Air enters the turbine room principally through louvres above the air washer shown in Fig. 5. This air, after being washed if necessary, passes down through the turbine room, picking up part of the heat lost by the steam apparatus. Ultimately it is drawn into the generator, picks up the heat representing the losses therein, and is discharged through a duct below the turbine room floor. This duct carries the heated air into the pipe gallery in which the stoker blowers are located. On its way to the blowers the air passes around the outside of the covering of all steam pipes, so that it also picks up the heat which these surrender to the surrounding atmosphere. In this way a large fraction of these unavoidable losses is conserved by being delivered to the boiler furnace, and just that much less heat need be supplied by fuel.

The electrical end of the plant also contains several unusual features. Notably, the generator leads are tied solidly to transformers which have a one to five ratio. These step the voltage up to 24,000, at which voltage all metering and switching is done. One of these transformers is shown to the left of the turbine room in Fig. 5 and also near the lower part of Fig. 13. There are three 8,333 k.v.a., single-phase water-cooled transformers per turbine.

FIG. 10 INTERIOR OF SCREEN HOUSE

A semi-diagrammatic layout of the complete electrical system is given in Fig. 12. The turbines will really be arranged in one straight row when all are installed, but the bus is actually built in the form of a loop, as shown in the figure. Section switches are provided between bus sections corresponding to each turbine and each of these sections is, in turn, divided into two approximately equal halves, thus localizing the effect of breakdowns to the greatest feasible extent. These section switches are non-automatic.

All switches are mounted in separate compartments on the floor immediately above the bus structures, as shown in Fig. 13. All machine, section and feeder switches are motor-operated, oil-break switches and are provided with lock type disconnecting switches on both sides, so that they can be completely killed for cleaning or repair. The doors on the oil switch and disconnect compartments are automatically locked by the closure of the oil switch.

On the floor above the switch gallery are located the storage battery (for auxiliaries, switches, and signals), the switchboard, and small electrical auxiliaries such as a balancer set, battery charging boosters, and a control motor-generator. The switchboard is located at what will ultimately be the center of length of the power house and is built in two sections arranged back to back, as shown in Fig. 13. The section facing the turbine room contains only the control switches and indicating instruments, while the rear board contains the integrating instruments and protective relays. All electrical equipment is controlled from this gallery, with the exception of the motor driven auxiliaries. The remote control switches for these are located to the left of the turbine foundation, as seen in Fig. 13. The auxiliary starting push buttons and the controllers for variable speed motors are located at the most convenient points, generally near the motor. The switch gear current for all apparatus handled from the main switchboard is supplied by the 25 kw. control motor-generator set mentioned above. This floats on a section of the storage battery, thus insuring continuity of service on the control.

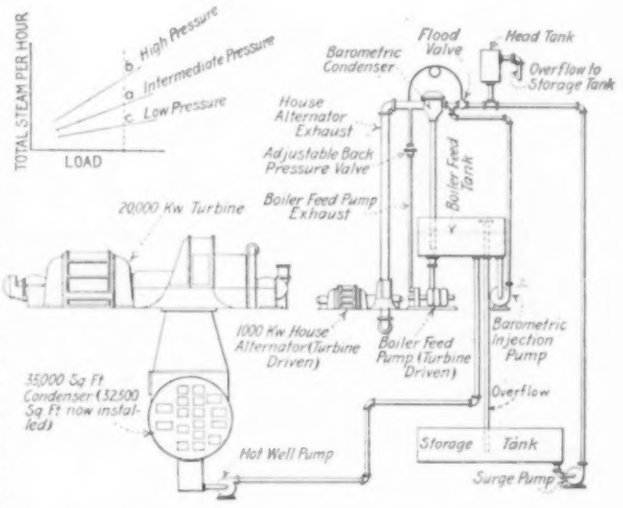

FIG. 11 ARRANGEMENT OF THE HOUSE SERVICE SYSTEM

In explanation of the fact that, as shown in Fig. 12, no reactances have been installed and that the design therefore diverges from recent practice, attention should be called to the fact that the inherent reactance of the apparatus connected to the busses, together with the nature of the bus structure, is such that no protective reactances are deemed necessary. Provision has been made in the bus structure for the installation of additional supporting insulators opposite to those already in place when the connected generator capacity becomes such as to demand additional rigidity, also for feeder reactances if changes of distribution methods later should make these desirable.

Each turbine is normally excited at 250 volts by an exciter mounted on an extension of the generator shaft. In ordinary operation, the armature of the exciter supplies the generator field directly, voltage control being effected by a motor-driven dial switch on the exciter's field rheostat. The gen-

erator field may also be excited from the direct-current house service bus, in which case it is controlled by a motor operated rheostat in the main field circuit. The house service direct-current bus which supplies the direct-current auxiliaries and the emergency excitation, is supplied by motor generator sets, one to each station unit. The station storage battery also floats on this bus insuring continuity of service.

The use of alternating current motors for some auxiliaries and direct current for others is determined by the following considerations: *a*. For any constant speed motor, alternating current was chosen. *b*. For speed variation of small range (as for the circulating pump where 15 per cent variation is sufficient), or of only occasional use (as the crane motors), alternating current motors with external resistances were chosen; except for the dry vacuum pumps, where the desirability of being able to start up from a condition of

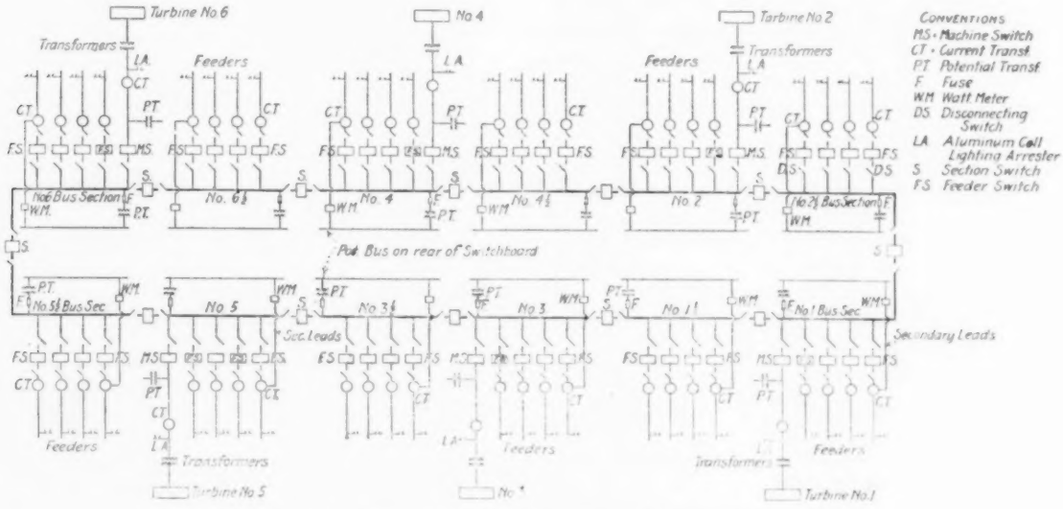

FIG. 12 LAYOUT OF THE COMPLETE ELECTRICAL SYSTEM

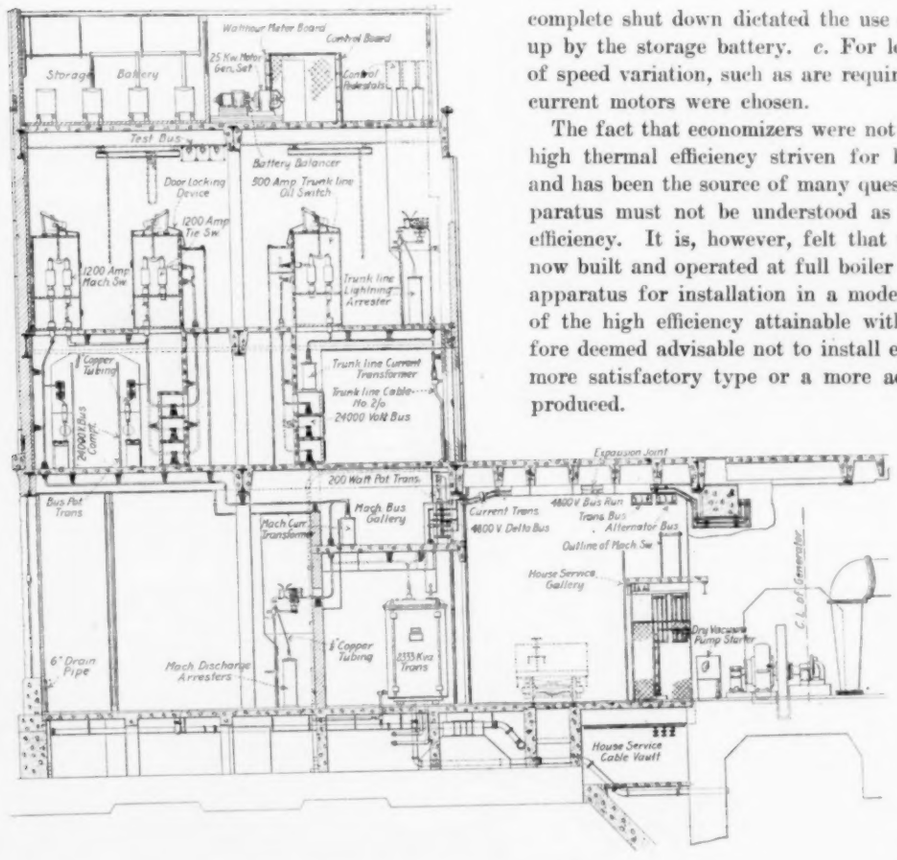

FIG. 13 SECTIONAL VIEW THROUGH SWITCH HOUSE

complete shut down dictated the use of a direct current motor backed up by the storage battery. *c*. For long ranges and exact adjustment of speed variation, such as are required by stokers and blowers, direct current motors were chosen.

The fact that economizers were not installed in this plant despite the high thermal efficiency striven for has caused considerable comment and has been the source of many questions. Failure to install such apparatus must not be understood as a declaration of disbelief in its efficiency. It is, however, felt that the cast iron tube economizer as now built and operated at full boiler pressure is not a proper piece of apparatus for installation in a modern high pressure plant. Because of the high efficiency attainable with the large boilers, it was therefore deemed advisable not to install economizers until such a time as a more satisfactory type or a more acceptable method of operating is produced.

NOTE. Since the presentation of this paper, the Connors Creek plant has been put in operation and thoroughly tested out. Small troubles, such as would be expected with a new and radical design of this sort have, of course, been met, but, on the whole, the plant has operated very smoothly indeed.

The operating record shows that the thermal efficiency of the plant will be about as high as was expected when it was laid down. The coal consumption per kilowatt-hour during the first two months of operation was considerably lower than the record performance of the Delray plant, and it is natural to expect that it will be further decreased as the operating methods are improved.

BOILER FAILURES AND WHAT THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS IS DOING TO PREVENT THEM

BY E. R. FISH, ST. LOUIS, MO.

Member of the Society

THE use of vessels for generating steam under pressure is as old as the use of steam itself, and for just that length of time there have been boiler failures of different kinds. The art and practice of boiler making has steadily progressed from its beginning with low pressures, imperfect and ill-suited materials, mediocre workmanship and unscientific designs, but it is, however, still far from perfect as is evidenced by continued diasters of varying magnitude.

Much has been written on the subject of boiler explosions, but Prof. R. H. Thurston was the first to put the matter in concrete and scientific shape, and I am quoting freely from his writings, for he expresses the conditions far better than I possibly can. Professor Thurston enunciated the following principles:

a Steam boiler explosions are always the result of well understood causes and are only mysterious, in any case, in the sense that available evidence may not point out with certainty which of the various well known causes may have operated in the given case.

b Boiler explosions are always preventable; it is always practicable to so design, construct and manage steam boilers that there shall be absolutely no danger of explosion.

c The proper course on the part of the designer, builder and user is to first use forms of boilers wherever practicable, such that, as Col. John Stevens said a century ago, "their explosion shall not be dangerous even should overpressure occur"; the same idea was enunciated by Fairbairn. Insure next good and well tested materials, good and well inspected workmanship and uniformly skillful management by men of experience and reliability. Regular, effective and thorough inspection constitutes a part of good management and an essential part.

He further stated that "steam boilers are magazines of energy forcibly restraining explosive powers of such magnitude as few can compute or realize." As an example, I select from one of the tables he calculated covering a number of different types and sizes of boilers, a return tubular boiler of 60 h.p. working at 75 lb. pressure. The weight of such a boiler is approximately 9500 lb., the water it contains weighs 8200 lb. and the steam 21 lb. The available stored energy in the water is 50,008,790 ft. lb. and in the steam 1,022,730 ft. lb., or a total of 51,031,520 ft. lb., which is sufficient to project the boiler only to a height of 5372 ft. with an initial velocity of 588 ft. per sec.

Quoting further from Thurston:

It is seen that the energy stored in the steam boiler, in the form of heat, is mainly that of the heated water and comparatively little resides in the steam; so that it follows that a boiler well filled with water is vastly more dangerous in case it explodes, than if exploded as a consequence of low water. The steam itself is quite incompetent to perform the work of the very destructive explosions often observed, and where the boiler is thrown, with the violence of a projectile from a piece of ordnance, against surrounding objects or hundreds of feet into the air; for it must be remembered that in no case can more than a small fraction of this computed total energy be actually applied to the propulsion of the boiler. Most of it is inevitably wasted in other directions.

Comparing the energy of water and of steam in the steam boiler with that of gunpowder, as used in ordnance, it has been found that at high pressures the former become possible rivals of the latter. Taking the value of gunpowder at what the writer would consider a fair figure, 250,000 ft. lb. per lb., it is seen that a cubic foot of heated water, under a pressure of 60 or 70 lb. per sq. in., has about the same energy as one pound of gunpowder.

A powder magazine under one of our modern tall buildings, or under the sidewalk in Broadway, would be objected to very seriously by citizens compelled to make that great thoroughfare their daily walk, and if compelled to submit to its threatening presence, they would certainly insist upon the most stringent precautions being taken to insure safety against the liberation of its stored energy; but scores of steam boilers are so concealed and the laws of the city and of the State relating to their care are of the most lax and ineffective character.

Good design, good workmanship and good management have been said to be the three safeguards against destructive explosion. Good design, in any case, presupposes a well read, well educated and an experienced designer.

Good construction involves the selection of good materials and their proper use in building the boiler. This means, in turn, iron, or more commonly, steel in which ductility and toughness, rather than simple strength, are the special characteristics, and perfect uniformity in the several sheets of which the boiler is made up. Good construction involves the exact production of the required forms of parts and their riveting, or welding, together without loss of strength or introduction of stress. Good management means the regular and moderate firing of the furnace, constant inspection of the parts liable to corrosion or other injury, and maintenance in as perfect condition as possible by constant repairing of injured parts. It also includes the periodical inspection and test which now constitute the basis of insurance.

All failures are due to preventable and familiar causes that might be evaded by the designer, the builder or the operator. Whatever may be said of the many known and possibly unknown, the certain, the probable and the possible causes, mysterious or otherwise, of boiler explosions, it is certain that a boiler is practically safe for all time in the hands of a good engineer or fireman if originally well designed, well constructed and properly set. Its safety is assured by a correct system of inspection either on the part of the person in charge or of the official inspector, or, more usually and properly, of both working together honestly and in good faith. A correct system of inspection is the check upon every defect of design, of construction and of operation.

Inspection should be made at fixed regular intervals and should be conducted by an inspector experienced, reliable, of good judgment and absolutely conscientious. He should be accompanied in his inspection by the responsible man in charge of the boiler, and his examination should be deliberate, thorough and unimpeachable. The time should be so chosen that the work may be done satisfactorily, without haste or interruption. The method should be—careful examination by eye, hand and a light hammer, of every sheet, stay, brace, tube and rivet in the boiler. Defective riveting, the most common defect, should be looked for in every seam. Cracks in the seam, under the lap, are not unusual and are very difficult to find in many cases, but if undiscovered, they will be likely to prove disastrous. Corroded sheets and tubes, bad bracing, sediment and incrustation are all promptly de-

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tected and readily visible in any case. The thickness of thinned sheets, corroded on either side, or of blisters, may be judged by the action of the sheet under the hammer, and many defects, entirely invisible, are found by the experienced inspector by the hand and ear, reinforced by the use of this tool. It is evident that no inspection worthy of the name can be made unless with the boiler dry and empty, and accessible in every part. A design which does not permit this should be condemned offhand. Tests by hydraulic pressure, though useful, and in the judgment of the writer indispensable, cannot be, in any case, allowed to supersede the real inspection above outlined.

Whatever may be the fact, however, more perfect and effective methods of inspection must be legally introduced and well enforced before the steam boiler explosion can become extinct, with its resultant destruction each year of hundreds of lives and millions of dollars' worth of property. With such methods, universally practiced, steam boiler explosions will become unknown.

Written some 25 years ago, the advice thus given has been to a considerable extent followed, but there still remains much to be accomplished, particularly as regards proper state legislation.

There are approximately 1500 boiler explosions of varying intensities each year in this country, averaging 400 or 500 persons killed, 700 or 800 injured and many hundreds of thousands of dollars damage to property.

General methods for proportioning the several dimensions of the boiler structure have long been known and in general use, although modified from time to time by tacit consent as experience or new and improved materials and methods became available. These rules, if such they may be called, have varied greatly in details in various places, depending on the personal opinions and experiences of the man or men formulating them. It can readily be appreciated that with the great multiplication of such regulations in the past few years, manufacturers who do a wide interstate business have become confronted with a condition that is hard to meet. In addition there are the infinite variety of private specifications written by consulting engineers and others, that differ through exceedingly wide limits and which represent the judgment of one or at best a very few men. Under such conditions it is impossible to manufacture economically, for a boiler built to meet the requirements of one place may be rejected in some other locality.

In 1905 there occurred in Brockton, Mass., an explosion which caused $250,000 damage, the loss of 58 lives and 117 injuries. This aroused the State of Massachusetts to action, the result of which was the enactment of legislation providing for the licensing of engineers and firemen, and governing the construction, installation and inspection of boilers. A Board of Boiler Rules was created for the purpose of formulating construction standards; its labors resulted in the most complete, scientific and logical set of regulations that had ever been promulgated. These rules have been quite extensively copied all over the country.

However, the need of regulations that would represent the consensus of the opinion and experience of the whole country rather than that of any one locality, prompted Col. E. D. Meier, when president of the American Society of Mechanical Engineers, to appoint a committee to formulate such rules. After more than 3½ years' real work, including numerous and lengthy discussions, often heated and acrimonious, the A. S. M. E. Boiler Code, as it is popularly known, has resulted and represents as nearly as it is humanly possible what those who are best qualified to know believe to be the best boiler practice.

There is much in the Code that is absolutely new. Probably only such a body as formulated it could have prevailed upon certain antagonistic manufacturers of materials and fixtures to get together and agree on uniform specifications for their products. This is notably the case with safety valve manufacturers and tube makers; possibly the most remarkable achievement is the very definite regulations, most of which are entirely new, relative to safety valves.

The Code covers the construction of stationary boilers and allowable working pressures, and is divided into two parts. Part I, Section I, applies to new installations of power boilers; Part I, Section II, to new installations of heating boilers, and Part II applies to existing installations of all boilers. Obviously it is necessary to extend considerable leniency to boilers now in use, as to do otherwise would work a tremendous hardship on thousands of steam users.

In brief, the subjects covered by the Code are indicated by the sub-headings: *Selection of Materials,* wherein is specified what grades of materials may be used for various parts and places in the boiler; *Ultimate Strength of Material Used in Computing Joints,* in which unit stresses forming the basis of calculations are given; *Minimum Thicknesses of Plates and Tubes,* to prevent "skinning" the job; *Specifications* for boiler plate steel, rivet steel, staybolt steel, steel bars, steel castings, gray iron castings, rivet iron, staybolt iron, and tubes, all of course as pertaining specifically to boiler practice. Then follow under the heading: *Construction and Maximum Allowable Working Pressures for Power Boilers,* detailed rules for determining strengths and efficiencies of joints, etc., with examples and stating in what way much of the work shall be executed. Then come the subjects of *Manholes, Washout Holes, Threaded Openings, Safety Valves, Water and Steam Gauges,* and *Fittings and Appliances,* wherein are given methods for determining quantities, sizes, limitations as to location and kind of metals to be used. Some general regulations as to settings to promote safety in operation, as to how hydrostatic tests shall be made and finally the method of stamping to show conformity of the structure with the Code, are given.

The rules covering boilers used exclusively for heating cover substantially the same ground, but are much more brief.

In Part II appear the rules for determining the working pressure to be allowed on existing boilers, and what fittings must be provided and how applied. On boilers now in service, will be required many minor changes if the Code is followed, but there are none but what should be made in any event. This section also provides for the gradual amortization, to use a commercial and financial term, of boilers now in use and not built to the Society standard.

An *Appendix* gives full examples of various riveted joints and methods of calculation, as well as of braced and stayed surfaces, method of computing safety valves, also how fusible plugs should be located in the various types of boilers, and standard dimensions for flanged pipe fittings.

In my opinion it is amply sufficient for any one desiring to purchase a boiler plant, after having determined the working pressure and the size and type of units required, to merely specify that the boiler is to be built according to the A. S. M. E. Code. By so doing he will get from any reputable

manufacturer a well constructed structure entirely suited to its purpose. If preferred, the material and workmanship may be checked by some inspecting agency. From a manufacturer's standpoint, my plea is for country-wide uniformity of requirements, and this the general adoption of the Code makes possible. It couples together economical manufacturing with the all-pervading cry of Safety First.

In compiling this Code, therefore, the American Society of Mechanical Engineers has provided a substantial basis for boiler specifications and legislation. The Society cannot go further than to say that, in the estimation of those best qualified to judge, these are the most rational rules and regulations for boiler construction. The enforcement of the rules must be left to the activities of other agencies. A very considerable number of the boiler manufacturers of the country have already gotten together on two occasions, first, on September 19, 1914, and again on March 29, 1915, with the avowed purpose of furthering the universal use of the Society's Code, and at the last named meeting, the Code was unanimously endorsed and steps taken to launch a movement to prevail upon the various States to enact legislation putting it into legal effect. It seems impracticable to do this in any other way. The State of Ohio has already made this enactment and the State of Wisconsin will in all probability do so, to be effective January 1, 1916.

Educational movements will have to be started in various other States in order to be at all sure of any success at future state legislative sessions. Last fall it was hoped that the Code would be completed in time to present the matter definitely to the numerous legislatures that were in session early this year, but that proved to be impracticable. One of the forms of education is to persuade those who have positive ideas of their own that such should be subservient to this broader consensus of opinion, and only by so doing can country-wide uniformity, so exceedingly desirable, be attained. Already there are some objections raised to minor points, which indicate that there are difficulties to be overcome in arriving at the desired end.

That this effort of the Society in bringing out the Code will help tremendously, primarily in bettering boiler construction generally, and secondarily, in smoothing the way of manufacturers, thus redounding to the financial benefit of the user, needs no further argument. The Code has already attained such momentum as to make it a powerful factor.

It can truly be said that what the American Society of Mechanical Engineers is doing to prevent boiler failures in this country is something which could have been done by no other organization and that the effect will be far reaching and long continued.

NOTE. Since this paper was written, an action has been taken at the annual convention of the American Boiler Manufacturers Association, June 22, to start a nation-wide movement including every interest in any way affected. This has resulted in the formation of the American Uniform Boiler Law Society, whose function will be to promote the adoption of the A. S. M. E. Boiler Code by legislative bodies.

TURBINES VS. ENGINES IN UNITS OF SMALL CAPACITIES

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THE term "units of small capacities" as used herein is intended to include steam turbines and engines of less than 500 h.p. capacity. The paper will necessarily deal largely with the prime movers of auxiliary apparatus in power plants, since the tendency of the times in all industries, and particularly in central stations, is toward the concentration of power in a few units of large size and uniform capacity as opposed to a multiplicity of small units of different capacities. However, there is a wide field for power application where the steam-operated prime movers are of relatively small size, and where transmitted or central station energy is not able to successfully compete; and it is intended to discuss the type of apparatus best suited in these cases, as well as the type of apparatus which it is advisable to employ for auxiliary units in large plants or central stations.

There are certain definite fields where the small turbine is of conceded superiority, and other fields wherein the engine must hold sway. The desirability of the one as compared with the other is largely determined by the following factors, which govern the adaptability, cost and economy of the equpiment to be installed for any given service:

a *Speed conditions and limitations* involving consideration of maximum or minimum permissible speed, and whether the driven apparatus is of the constant or variable speed class

b *Steam pressure and temperature conditions* involving consideration of initial and final pressures, and superheat, if any

c *Power capacity of apparatus*

d *Relative space requirements of turbine and engine units* involving consideration of available room, character of power house construction and cost of foundation or other supporting structure

e *Use or application, if any, of the exhaust steam* for feed water heating, steam heating or process purposes

f *Available cooling water supply,* if the turbine or engine is to run condensing, involving also consideration of the temperature of the water and whether it must be artificially cooled and re-circulated

g *Operating conditions* including consideration of attendance, oiling, starting and stopping, vibration, noise, etc.

h *Relative cost of complete installations* including necessary foundations, piping and condenser equipment, if any

As to the practicability of the small turbine, it may be said that not until about 20 years ago was any really practicable apparatus of this kind developed, and even up to ten years ago the turbine was looked upon mainly as an experiment. The last few years have witnessed, however, the practical perfection of this type of prime mover in sizes

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as large as 50,000 h.p., with units of 30,000 h.p. quite common in large central stations. They have also seen quite as much good work done in the perfection of small turbine units as in the development of very large ones, and the turbine in all sizes is quite as well developed to-day as is the steam engine after more than one hundred years of constant effort and improvement.

The writer has no intention of reflecting on the great work done by steam engineers in the development of the reciprocating steam engine. The work of these men, which was the harder task, accelerated the evolution of the turbine, the science of thermodynamics underlying the final development and perfection of both.

The present day builders of reciprocating engines are able to report further progress, however, and as a result of their recent efforts there have appeared the rejuvenated poppet valve engine adapted to the use of highly superheated steam; the unaflow or parallel-flow engine, also with poppet type valves, in which cylinder condensation is reduced by causing the steam to travel in one direction only; and the small self-contained power plant, or "locomobile," consisting of a steam engine mounted upon a tubular boiler and operating at high superheat. All these engines are adaptations from European practice, where, owing to the high cost of fuel and the relatively larger number of technically-trained operators, they have found high favor.

Mention also should be made of the power plant formerly used in the White motor car, which, owing to its high steam pressure, superheat and speed, gave remarkably low steam consumptions. In a series of tests, reported in the Transactions of the Society, Vol. 28, pages 598-9, the average steam consumption was 12.7 lb. per delivered h.p.-hr., operating at 850 r.p.m.; steam at 275 lb.; superheat, 349 deg. fahr.; noncondensing exhaust.

Tests have recently been reported by the manufacturers which give the steam consumption of a 115 h.p. Buckeye-mobile, running at 248 r.p.m.; steam at 210 lb.; initial superheat, 171 deg. fahr.; noncondensing, as 13.3 lb. per i.h.p.hr. This unit produced an i.h.p.hr. on 1.33 lb. of coal having a calorific value of 14,500 B.t.u. per lb. A 169 h.p. unit running at 200 r.p.m.; steam at 209 lb.; initial superheat, 218 deg. fahr.; vacuum 25.7 in., showed a water rate of 9.2 lb. per i.h.p.-hr. The coal consumption, using fuel with a calorific value of 14,209 B.t.u. per lb., was 1.08 lb. per i.h.p.-hr.

While there is the possibility that the use of a shell type of boiler for high pressures will be considered by many engineers as unwise, especially for installation in densely populated centers, it is still too early to predict what the future of this apparatus is to be. However, as indicating what may be accomplished in fuel economy in a well-designed plant of small size, the results reported are interesting.

SPEED LIMITATIONS

The question of speed limitations is of first importance in selecting the type of prime mover. Since high peripheral velocities are necessary in order to efficiently utilize the energy of a steam jet in the turbine type of apparatus, the latter shows its lowest water rate when running at a constant high speed. Where the character of the service is such as to require speed variation or reversal in direction, or where the speed is necessarily low, the turbine is unsuited and the engine is much better adapted.

In engine installations, the minimum permissible speed has an important bearing on the question of operation. If an engine is run at very high speeds, operating troubles are sure to be numerous, the upkeep excessive and the service unsatisfactory. The lack of driven apparatus designed to run efficiently at speeds consistent with high turbine economy has, in the past, frequently dictated the use of engine prime movers for many kinds of work.

Although speed reduction gears are by no means new, having been used with the turbine almost from the beginning of its commercial development, improvements in high speed gearing, as well as in the manufacture of high speed direct connected generators, blowers and pumps, running at 3,000 r.p.m. and above, have greatly increased the possibilities for turbine installations. Direct current generators as small as 10 kw. capacity, and 60 cycle alternators of capacities as low as 150 kw., designed for gear drive, are now offered. The manufacturers claim for these machines that the increased efficiency of the higher speed turbine, together with the saving effected in the generator construction by reason of the slower speed permissible in the driven end, justify the expense and complication which the gears introduce.

While many may still prefer a direct connection, the increasing popularity of the gear drive, especially for direct current generators, blowers and pumps, would seem to indicate that the gears are here to stay, and that when properly constructed and installed, there is no valid objection to their use.

For power station work, where some of the auxiliaries are usually motor driven, the exhaust steam can be entirely condensed in the feedwater heater, and the water rate of the steam driven auxiliaries is not a limiting factor, while reliability, accessibility, low maintenance and labor costs are of more vital importance. Power station designers have always exhibited, therefore, a strong preference for turbo-auxiliary units, and there is now a decided tendency toward geared installations.

The selection of turbines for auxiliaries is largely influenced by the high speed at which small engine units are run, which makes it exceedingly difficult to keep them in continuous service, and almost impossible to secure smooth, quiet operation. Such reciprocating units require close attention, and must be shut down, overhauled and adjusted at frequent intervals; the maintenance is high and serious breakdowns are by no means rare. An accident to a circulating or hot well pump, for example, usually necessitates a shutdown of the main generator, with consequent loss of production, and often, in the case of a public utilities plant, loss of prestige and the incurrence of public ill-will. In all central stations, therefore, where the main units are few in number and of large size, high economy as compared with continuous operation becomes relatively unimportant, and turbine driven circulating hot well and boiler feed pumps are almost invariably used.

Motor driven exciters are largely used in alternating current stations, a steam driven exciter being provided for starting up and as a reserve unit. Here also, since the steam driven set is idle a large part of the time, high economy is not so important and the saving of floor space and elimination of vibration will often decide the question in favor of the turbine. In fact, in all direct current generator sets of 50 kw. capacity or less, the high speed necessary for the engine generally makes it undesirable.

For driving fans of large capacity at low pressures, say less than 1½ in. of water, for induced draft, hot air heating and ventilating systems and the like, the engine seems best suited. Fans built for this service ordinarily run at less than 200 r.p.m. and are usually of the paddle-wheel type, which is better suited to these conditions than the multiblade high speed fan. In induced draft work, load fluctuation may require frequent changes in speed, the engine being under the control of a throttling regulator which is automatically actuated by a change of steam pressure. These conditions are quite unfavorable to turbine economy, and a suitably designed and well-constructed engine will give more satisfactory results.

In like manner, where we consider the pumping of large quantities of water against low variable heads (conditions which are encountered in drainage or sewage pumping stations) the turbine must yield place to its rival. These pumps may have single runners of large diameters or may be of the multi-impeller type, but in either case, the speed is below the economical turbine range, even when a gear is used, while the engine, at low speed, has an opportunity to make its best showing, steam economy and operating troubles considered. Where the lift is variable, speed changes are required and the engine is almost always more suitable.

As a typical installation may be mentioned the four 76 in. centrifugal pumps for the Placquemine and Jefferson Drainage District of Louisiana. Here the pump speed varies from 50 r.p.m. for 1 ft. head and 135,000 gal. per min. to 115 r.p.m. for 13 ft. head and 90,000 gal. per min. The best duty was 92,600,000 ft. lb. per 1000 lb. of dry steam, corresponding to 21.4 lb. of steam per water horse-power obtained at 87 r.p.m. for 7 ft. head and 130,000 gal. per min. The prime mover is a compound engine operating at 170 lb. steam and 25.7 in. vacuum.

The use of underfeed stokers operating under heavy forced draft and capable of developing high boiler ratings, has become quite common as a means of reducing fixed charges and boiler banking losses in railway and lighting plants, and particularly those maintained as standbys to hydro-electric stations. These furnaces often carry in the air duct pressures as high as 6 in. of water, and the high speed multiblade fan makes the better installation, particularly where one fan serves several boilers, as the blower units frequently become excessively large when run slower than 400 r.p.m. At this speed the engine drive is an uncertain and expensive proposition.

Furthermore, as such stokers at best are capable of only from one-quarter to one-third their maximum capacity under natural draft conditions, a blower breakdown under a peak load is a serious matter and the ability of the turbine to stand up under the conditions imposed justly entitles it to the preference which it is accorded.

For driving directly connected alternators, a frequency of 25 cycles fixes the maximum speed at 1500 r.p.m., which is too low for the best turbine performance. In 60-cycle apparatus, where a speed of 3600 r.p.m. is possible, the turbine shows to better advantage.

STEAM PRESSURE AND TEMPERATURE CONDITIONS

The highly economical steam turbine must necessarily be operated condensing, but, as previously pointed out, there are many cases where high steam economy is not the most important consideration and the noncondensing turbine often finds favor over the steam engine. One of the large heat losses incurred in the steam engine is that due to cylinder condensation and one of the common methods of reducing it to limit the range of temperature through which the steam is allowed to work in a single cylinder. For this reason simple engines are better adapted for low steam pressures, while compound and triple expansion engines are advisable for high pressure and high temperature ranges, particularly if the load is uniform.

The engine as a rule develops mechanical troubles with high superheat, especially where the steam valves have much travel under unbalanced pressures. The consensus of opinion seems to be that for slide or gridiron valve gears, a temperature of 400 deg. fahr. to 425 deg. fahr. should not be exceeded, while the best point for the Corliss type engine will be found below 450 deg. fahr. Above these limits lubrication is unsatisfactory and distortion of the parts is apt to give trouble.

In European practice, superheating is much more common than in this country, a superheater being considered quite as indispensable as a feed water heater and the poppet valve engine, which was first perfected abroad, is accordingly better suited for high superheat conditions than the type of gear commonly used in American engines.

A difficulty sometimes encountered with engines using high superheat is the warping of the cylinder, the curvature being caused by the higher temperature which prevails in the metal next to the steam chest. Precaution is taken by some builders to avoid such trouble by leading the steam by two independent pipes, entirely separate from the cylinder barrel, from the throttle directly to the steam valves.

For power plant auxiliaries, it would appear that turbines which experience little difficulty from high temperatures will be more and more widely adopted while engines will be less commonly used, especially so as steam pressures and superheats are constantly increasing, it being not unusual for new plants to be designed to carry from 200 to 225 lb. with superheats of 150 deg. fahr. and over. With steam engines running under high vacua, above 27 in., the great volume of steam to be handled increases the size of the engine cylinders, and the size of ports through which the steam must pass, to such an extent as to make the engines very expensive if not of impracticable construction. The cost of an engine which would permit of complete expansion to such a terminal pressure, together with the increase in cylinder condensation, due to the greater range of temperature, would make the high vacuum undesirable.

The turbine, on the other hand, can be designed to operate on very low terminal pressures with comparatively slight increase of cost; its action as a heat machine is such that a greater expansion can be utilized and the economy is greatly improved by any increase in vacuum. When run non-condensing, as is well known, the turbine is less economical than the non-condensing engine.

In plants where the exhaust is atmospheric and cannot be applied to any useful purpose, the engine best fills the conditions, provided space is available and the speed may be made low enough to ensure smooth, quiet running. Such an application is found in direct current generator sets of 100 kw. capacity and larger, in hotels, office buildings and hospitals, where the exhaust is used for steam heating in winter and must be wasted for several months in the year.

POWER CAPACITY OF APPARATUS

The lower cost of large turbine units and the greater reliability of this kind of apparatus in regular service, coupled with the smaller space taken up by turbines as compared with engines, has practically put the engine out of the running as far as large power plants are concerned. Where 60-cycle apparatus is installed and condensing units are used, the engine has no field beyond the 500 kw. mark, while with direct connected 25-cycle apparatus, the engine must stop beyond the 1000 kw. limit, and with the perfection of high speed reduction gears, it is doubtful if 25-cycle engine driven generators can compete with turbine apparatus of even 500 kw. capacity. The reduction gear is also rapidly driving the engine from the direct current field in units of all sizes, above say 200 kw. capacity. At the same time, elimination of operating troubles by the use of direct connected turbines for exciter purposes is fast causing the turbine to supplant the engine for this service.

In the case of non-condensing units where moderate speeds are required, the engine must continue to hold the field, though special conditions may make the non-condensing turbine a factor to be considered. In this connection, one installation might be mentioned where a belted turbine of 750 h.p. capacity, running at 1500 r.p.m., is used for driving the constant speed shafting of a paper mill, it being contended that the greater uniformity in rotative speed secured by the turbine results in fewer breaks and a more satisfactory product. In this case, the exhaust steam is, of course, utilized in the dryers of the machine, and the variable speed power is supplied by direct current motors.

RELATIVE SPACE REQUIREMENTS

Owing to the freedom from reciprocating motion, the foundations required for turbines are of small size and light weight, there being little vibration to be absorbed when the alignment and balancing are well done. The small sizes can be safely operated on floors of usual construction, designed for the ordinary floor loads. There is no difficulty experienced with the transmission of vibration to the structural members of the building or to the piping system.

The small space required for the installation of a turbine gives it an advantage in water works plants, operating against moderately high heads. The vertical triple expansion engine, which was formerly used almost exclusively for such work, requires a strong massive substructure to absorb the shock and distribute the weight, and a deep pit to accommodate the water end. Where foundation or other construction difficulties are encountered in this work, the cost may easily climb to a high figure, and in a case under the author's personal observation the additional cost of building, incidental to the use of the vertical triple engine, would have more than paid for a turbine driven centrifugal pump, while the fixed charges on the pump alone were more than four times the cost of the fuel required to run it to full capacity ten hours per day.

The turbine driven pump is not capable of showing on test the high duty of the vertical triple engine—which is one of the most economical steam engines—but the great difference in the first cost of installation often makes the turbo set decidedly preferable, especially when the saving in building is considered. A geared turbine unit to pump 100 million gallons per day against a 56 ft. head was recently installed in Ross Station, Pittsburgh; the pump speed was 350 r.p.m. and that of the turbine 3600 r.p.m. On the duty trial, with steam at 151 lb. and vacuum 28.38 in., the pump showed a performance, including power consumed by the auxiliaries, of 120.5 million ft. lb. per 1000 lb. of dry steam, corresponding to 16.44 lb. of steam per w.h.p.hr. Six similar sets, ranging in capacity from $6\frac{1}{2}$ to 30 million gallons per day capacity, are now in process of construction. The displacement of the reciprocating engine from a field where its superiority was formerly unquestioned, shows the substantial progress which has been made in the development of the steam turbine and the centrifugal pump.

For boiler feed pumps of more than 250 gal. per min. capacity, the turbine is often used, and on account of its small size, usually results in a neater and more compact layout. Where regulation by throttling is unnecessary, and the pumps run at or near capacity, the economy as compared with the direct acting type is good and can be better maintained. Valve renewal and packing troubles are avoided. The overload capacity of the centrifugal type, however, is small and the delivery of the pump must be proportioned to meet the maximum demand, not the average boiler horsepower requirements. In the smaller sizes, the cost of turbine units is high; where the load fluctuates widely and the speed must vary, the economy is poor and it is better to install reciprocating pumps.

In the modern plant containing large turbo-generator units, space limitations in the basement arrangements are an important consideration. With the high vacua carried, large volumes of water must be handled and the turbine drive for circulating, condensation and air removal pumps is in many cases the proper selection. Such condenser sets have a compact arrangement, especially when a single turbine is used to drive all the pumps, which greatly relieves the crowded condition that would otherwise obtain. As previously stated, they are also preferable as being more reliable.

The turbo-compressor supplying air to blast furnaces under pressures ranging from 20 lb. to 30 lb. has almost entirely supplanted the compound reciprocating blowing engine. One large concern formerly in this work has abandoned the construction of blowing engines and is now building turbine apparatus exclusively. For this service, the turbine may be run from 2500 to 4000 r.p.m., and there is a great saving of weight and space, as an engine of this type is six or eight times as heavy as the centrifugal blower, and consequently costs much more. It can be installed comfortably where the blowing engine would be out of the question, and in new installations the relative cost of building and foundation for the two types has a direct and important bearing.

UTILIZATION OF EXHAUST STEAM

The advantage of an oil-free exhaust is in many plants of considerable value, and especially so in manufacturing processes where steam is used, as there are many such opportunities for the utilization of low pressure steam if the oil has been eliminated. For the blocking of hats and in the treatment of other felt and textile products, absolutely clean steam is necessary. As heretofore mentioned, paper manufacturers have used turbines for driving the constant speed mechanism of paper machines in order to secure more uniform angular velocity, and to avoid among other things

trouble caused by oil accumulation in the drying rolls. The danger of oil deposits in high pressure steam boilers is well-known to all.

In chemical processes where steam is used for precipitation, as in the precipitation of magnesia, a small fraction of a grain of oil per gallon will often retard the process or cause the precipitate to be of an entirely different character from that obtained with oil-free steam. The separation of the oil in exhaust steam is never absolutely complete, and fatty constituents are especially apt to pass the separator.

AVAILABLE COOLING WATER SUPPLY

Where the available cooling water supply is limited and must be artificially cooled and recirculated, the cost of the cooling apparatus and the power required must be considered. The conditions will, perhaps, be best illustrated by an example: Assuming a turbine to run at 28 in. vacuum, and a temperature rise of the cooling water to within 10 deg. of that due to the vacuum, the circulation of 52 units by weight of cooling water for each unit of steam condensed will be necessary. In the case of the engine, which will operate at, say, 26 in. vacuum, other conditions remaining the same, there will be from 25 to 27 lb. of cooling water to be handled for each pound of steam condensed.

With a cooling pond returning water at 90 deg. to produce 27 in. vacuum in a turbine plant, the pumps must circulate 70 units of cooling water per lb. of steam, as against, say, 30 units required to produce 25 in. vacuum for the reciprocating engine.

OPERATING ADVANTAGES

From the operating point of view, the turbine possesses a great advantage in the simplicity of its construction, a factor which tends toward increased reliability and lower cost of maintenance. It can usually be more quickly started and loaded and, in operation, usually requires very much less attention than an engine unit of corresponding capacity. The lubricating arrangements are few in number and of simple design.

SUMMARY

Summarizing the foregoing, the fields of usefulness of the turbine and engine may be briefly stated to be:

APPLICABILITY OF TURBINES

1 *Direct connected units, operating condensing.* 60-cycle generators in all sizes, also 25-cycle generators above 1000 kw. capacity. (This paper is, however, not intended to deal with units of this size.)

Direct current generators in sizes up to 1000 kw. capacity, including exciter units of all sizes.

Centrifugal pumping machinery operating under substantially constant head and quantity conditions, and at moderately high head, say from 100 ft. up, depending upon the size of the unit.

Fans and blowers for delivering air at pressures from 1½ in. water column to 30 lb. per sq. in.

2 *Direct connected units, operating non-condensing* for all the above purposes, in those cases wherein steam economy is not the prime factor or where the exhaust steam can be completely utilized, and, in the latter case, particularly where oil free exhaust steam is desirable or essential.

3 *Geared units,* operating either condensing or non-condensing for all the above mentioned applications, and in addition, many others which would otherwise fall in the category of the steam engine, on account of the relatively slow speed of the apparatus to be driven.

APPLICABILITY OF ENGINES

1 *Non-condensing units, direct connected or belted* and used for driving:

Electric generators of all classes excepting exciter sets of small capacity, unless belted from the main engine.

Centrifugal pumping machinery, operating under variable head and quality conditions and at relatively low heads, say up to 100 ft., depending on the capacity of the unit.

Pumps and compressors for delivering water or gases in relatively small quantities and at relatively high pressures—in the case of pumps at pressures above 100 lb. per sq. in. and in the case of compressors at pressures from 1 lb. per sq. in. and above.

Fans and blowers (including induced draft fans) for handling air in variable quantities and at relatively low pressures, say not over 5 in. water column.

Line shafts of mills, where the driven apparatus is closely grouped and the load factor is good.

All apparatus requiring reversal in direction of rotation, as in hoisting engines and engines for traction purposes.

2 *Condensing units direct connected or belted,* for all the above purposes, particularly where the condensing water supply is limited, and where the water must be re-cooled and re-circulated.

SPRING MEETING PAPERS

IN the August issue of The Journal were published eight of the fourteen papers presented at the Spring Meeting held at Buffalo, June 22-25. The six remaining papers are here published in abstract, together with accounts of the discussions following them. Pamphlet copies of the complete papers without the discussion are available at the prices named in each case. Later, the papers which are to appear in Volume 37 of Transactions, may be had in pamphlet form with the discussion added.

SOME MECHANICAL FEATURES OF THE HYDRATION OF PORTLAND CEMENT AND THE MAKING OF CONCRETE AS REVEALED BY MICROSCOPIC STUDY

BY NATHAN C. JOHNSON, NEW YORK, N. Y.

Member of the Society

THE data embodied in this paper have been obtained in the course of a research conducted in the Sibley College laboratories at Cornell University. At the beginning of this research, it was believed that the obtaining of more effective hydration was a prime factor in the production of durable concrete. It has long been known experimentally that a set cement or concrete could be reground and a new set obtained on gaging with water;[1] but the extent to which unhydrated particles were present in the mass was first made visually evident, so far as the author is aware, by experiments conducted in 1911 in the Sibley College laboratories at Cornell.[2] In these experiments, neat cement briquettes were surfaced and polished, as in the metallography of steel; and the degree of hydration obtained in different ways was more or less accurately judged by the appearance of these polished sections, viewed through the microscope. Later this same procedure was extended to the examination of 1.3 standard sand mortars and still later to the examination of field concretes.

In a section of such 1.3 mortars photographed through the microscope (shown in Fig. 3), the grains of sand show as large as boulders and each has a well-defined shadow. Further, since only cement and a sand of known size were used in the briquette, the "rocks" between the boulders can be nothing other than cement; and this cement must be unhydrated, since its color by direct vision, varying from light green to dark brown, and its resistance to the cutting of the polishing powders used in preparing the surface to remove the portion known to be hydrated, both testify to the same fact.

Such disclosures as these were the basis on which was undertaken the original research by the author, of which this Industrial Fellowship research was a later outgrowth. Since such structures were found to be typical of all mortars examined, and since cement is the most variable constituent of any mortar or any concrete, it was argued that any process which would insure thorough hydration of the cement particles would render possible the use either of higher unit-stress values, or of less cement for present allowable values, with a most pronounced saving in either case, both in materials and money.

From the nature of this investigation, it is at once evident that progress would have to be measured by checking values obtained by the apparatus against those obtained with the same materials under standard conditions of mixing and moulding. In addition to the standard physical tests, it was also proposed to carry on a study of the microsections of each set of specimens broken. To facilitate such a comparative study, record books were prepared for both hand and machine mixes, the hand mixes being designated by the letter *H*, followed by a serial number, with results recorded below; and the same data for machine mixes, designated by the letter *P* (pneumatic) followed by a serial number, are placed on the opposite page. The position of the break of each briquette was recorded in the outline form. Although not recommended for general use, this has proved of very great benefit in reconciling otherwise discordant results.

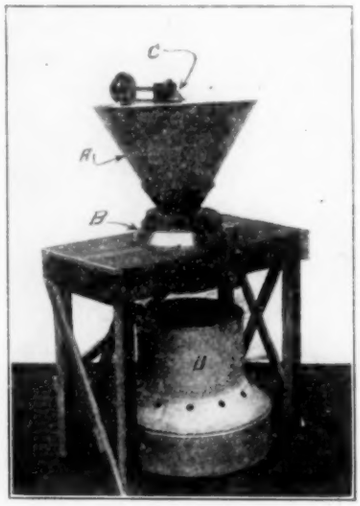

FIG. 1 IMPROVED FORM OF HYDRATOR

In Fig. 1 is shown one of the latest forms of hydrator used, which was believed to include the essential features of earlier types with the mechanically troublesome parts refined. In this photograph, *A* is a hopper, of sufficient size to contain a whole bag of cement; *B* is the atomizer, with the jets increased to forty-eight; *C* is a feed-device of coffee-mill type, driven by an electric motor; and *D* is a container with baffled exits. In spite of the fondest hopes and the most strenuous labor, this machine failed in its purpose, so far as really notable results are concerned. As the re-

[1] H. Sager & E. Cramer. Tonind Zeit, 1908, 32, 1746.

[2] Master's Thesis by N. C. Johnson, Cornell University, 1913.

Presented at the Spring Meeting of the Society, June 1915. Complete paper may be obtained without discussion; price 15 cents to members; 30 cents to non-members.

search proceeded, it was found that there originally had been considerable misapprehension as to the causes of faulty hydration, though not as to the extent or degree to which hydration commonly proceeds. The obtaining of better hydration is a very vital problem in the concrete industry, but to effect this on a commercial scale in the manner above outlined would be virtually impossible.

In regard to the microscopic investigation of cements, the process is, in general, similar to the microscopic examination of steels in metallography which has already proven of the greatest benefit to the steel industry. And as is done with specimens of steel, a single surface only is polished, the structure of this being developed by various reagents and observed through the microscope by reflected light. This renders possible the ready examination of composite materials, which would otherwise require the greatest patience and skill in their preparation.

The use of the photographic camera in connection with the microscope is but carrying the process a step further. On the microscope stage is mounted the polished and surfaced specimen, on which the light is concentrated by a bull's-eye lens. Above the eye-end of the microscope is mounted an ordinary photographic hand-camera with its proper lens removed; and on the ground glass at the top, for which may be substituted the sensitized plate, is projected the enlarged image of the structure of the specimen.

To ensure a thorough understanding of the evidences and effects of hydration of cement, it may be well to consider here the nature of the processes involved. The completed cement derives its usefulness from the hardening produced shortly after its admixture with water. There are, however, two stages in the passage from the semi-fluid state of gaged cement to that of the hardened. In the first stage the mass loses its plasticity and becomes more or less friable. In the second stage consolidation takes place, the mass increasing in hardness until a stony texture is obtained. These two stages are respectively distinguished as "setting" and "hardening."

It follows from the complex character of its composition, that the reactions involved in the setting and hardening of Portland cement are themselves complex. It seems probable, however, that the reactions of setting involve the formation of super-saturated solutions and the deposition therefrom of close-knitted, interlacing crystals of various substances, while the slower reactions of hardening consist partly in the formation of similar interlacing crystalline products, but more especially in the production of a colloidal glue, which is, probably colloidal calcium hydrosilicate, and its gradual desiccation.

The colloidal interpretation of the hardening of cements is due to Dr. Wm. Michaelis, Sr., who reasoned that a pure crystallization process, as in the case of plaster of paris, can never cause hydraulic hardening, for a conglomerate of crystals, however insoluble, cannot produce impermeability, since crystals have plane surfaces with voids between them which would admit water. On the other hand, hardened cement mortars are impermeable, so that crystallization alone could not confer on them their valuable properties. Dr. Michaelis found the medium responsible to be the colloidal silicates formed in the later stages of hydration.

Illustrative of such structures and also of the primary crystalline formations which have been indicated as being responsible for initial set, in Fig. 4 is shown a surface found in a concrete 34 years old, taken from a dock wall in New York harbor. The production of these crystals was quite unlooked for, being due to an unusually delicate etching of the piece of clinker after polishing. It should be remembered that this magnification is high (200 diameters) and that these crystals are formed on a single particle. Multiply such interlacing formations a few million times for each cu. yd. of concrete, and it might be expected that a very considerable degree of mechanical strength, quite comparable to the strength of initial set, would be developed. This particular crystalline formation is shown tentatively, however, in this connection, as its identity with those chemical compounds known to be responsible for initial set is as yet conjectural.

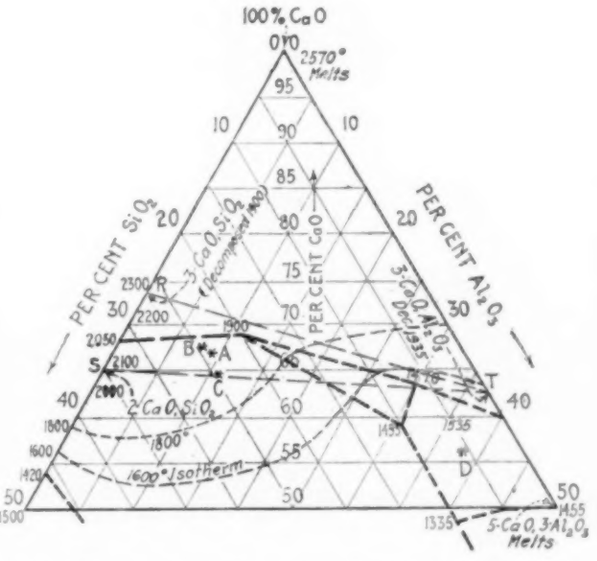

FIG. 2 PORTION OF THREE COMPONENT DIAGRAMS SHOWING PRODUCTS FORMED BY THE SINTERING OF VARYING PROPORTIONS OF PORTLAND CEMENT CONSTITUENTS

But since this attack on the problems of hydration is primarily engineering, rather than chemical, and must therefore necessarily be identified with a study of actual concretes, it becomes necessary to make sure that there is a proper understanding of the internal structure of concretes and of the relative size, position in the mass and importance of the several constituents.

In Fig. 5 is shown a sectioned concrete specimen, weighing about ten lb. and about 8-in. × 6-in. × 1.5-in. size. It should be noted that none of the aggregate, whether large or small, so far as they are here visible, are touching at any point; rather, they are widely dispersed. But void determinations on this stone were made when the pieces were in at least point contact. Evidently, the voids must have been increased enormously by this dispersion; and if similar dispersion obtains for the finer materials as well, it is no wonder that proportioning concretes on the basis of void determinations has become discredited.

In Fig. 6 is shown a section of the same surface, but at twice the magnification, in order that the finer materials may be made visible. So far as they can be seen, this same dispersion obtains. Carrying the magnification to four times

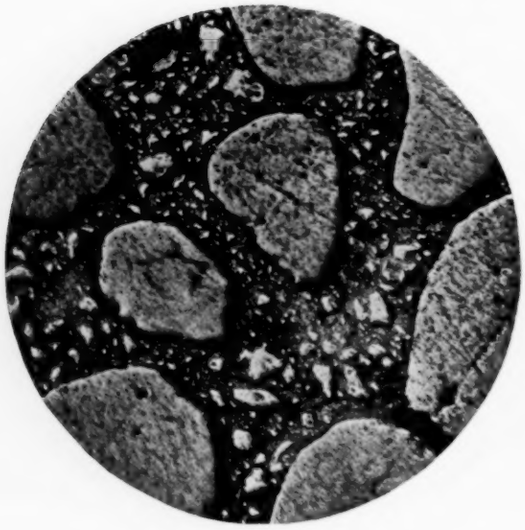

FIG. 3

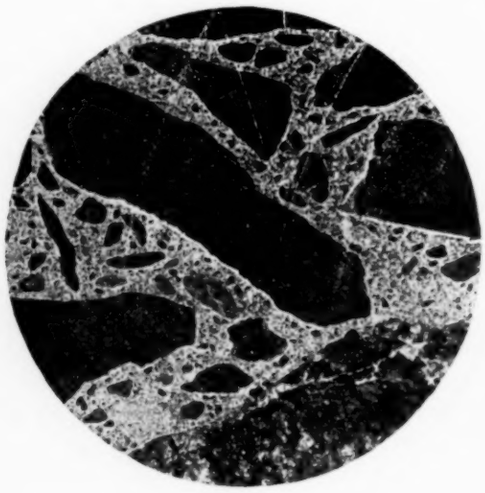

FIG. 6

FIG. 4

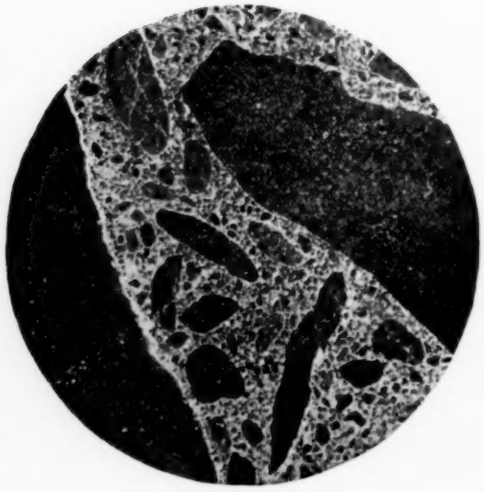

FIG. 7

FIG. 5

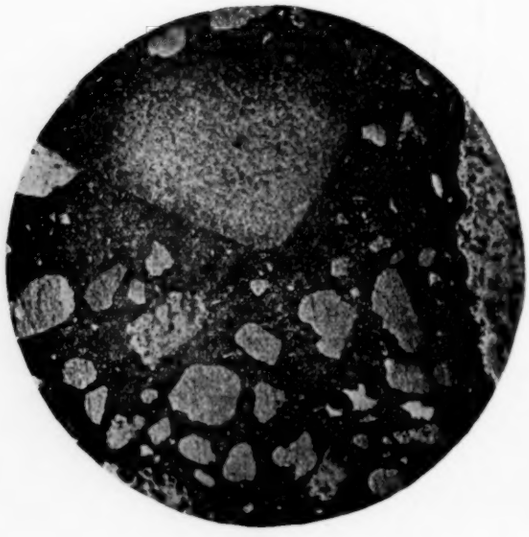

FIG. 8

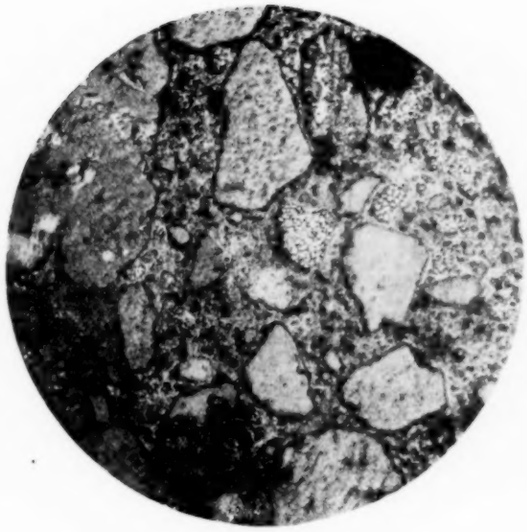

FIG. 9

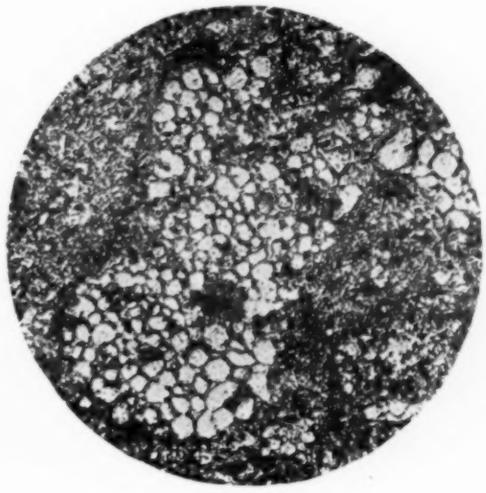

FIG. 12

FIG. 10

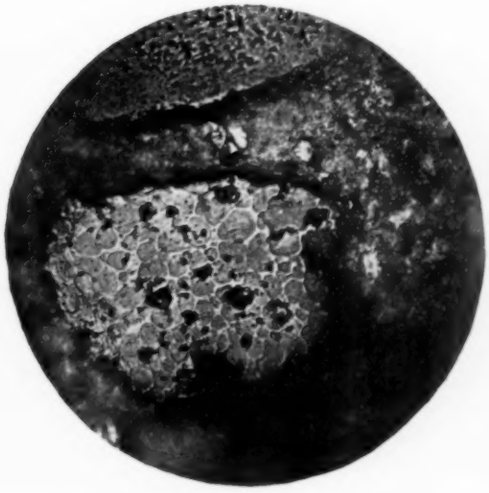

FIG. 13

FIG. 11

FIG. 14

the original, as in Fig. 7, this is made even more apparent; carrying it to 20 times the original, by means of photographing through the microscope, shows, as in Fig. 8, that all the aggregate in sight are sand grains of greater or less size, separated, as were the pieces of stone, and with still finer particles, too small to be sand, lying between them. Magnifying this same field to 60 times the actual diameter, as in Fig. 9, shows the sand grains in increased size, with a relative increase in their mutual dispersion; and a peculiar honeycomb structure on the smaller particles lying between them. Magnifying a portion of this same field 200 times, as in Fig. 10, this honeycomb structure is revealed more definitely. Since elimination of stone and sand from the known composition of the concrete indicates this honeycombed particle to be unhydrated cement and as similar formations occur profusely throughout all concretes, it becomes important to identify them beyond question and to study their functions in the rigid concrete mass, for, if they are unhydrated cement, lying inert, they have a most important bearing on the strengths of concrete.

The easiest identification is offered by comparison between the known and unknown material. In Fig. 11 is shown the structure of a neat cement briquette as viewed through the microscope at a magnification of 60 diameters. Comparing this with Fig. 9, which has the same degree of magnification, the identity of the two materials is strongly indicated. But to make this more certain, in Fig. 12 is shown the central portion of Fig. 11 but at a magnification of 200 diameters. Comparing this with Fig. 10, which shows the unknown material found in the concrete, their general characteristics are seen to be identical.

In an early paragraph of this paper, the proportion of material in standard sand mortar, which was then stated to be unhydrated cement, was seen to be large. In view of what has just been demonstrated, the truth of this statement is not difficult of proof. In Fig. 13 is shown a section of surface of a standard sand briquette, photographed at a magnification of 200 diameters. The structure of the cement group is here shown in an increased size, with the segmental edges of grains of standard sand, in its near neighborhood. Since in this briquette no materials other than cement, standard sand and water were used, the photograph serves as further proof of the identity of the questioned structure.

It may be objected, however, that this particle is too large to be a cement particle, and that if it were a group of cement particles, they would hydrate and fall apart. To answer this objection, recourse must be had to a consideration of the physical actions involved in wetting dry cement with water.

In Fig. 14 is shown at a magnification of 30 diameters the appearance of dry cement as it lies on a microscope slide. This cement film is very thin, a mere surface dust, but it will be seen that the distribution is very uniform. But if water is added, quite different conditions obtain. In Fig. 15, the water may be seen progressing across the field. Where the water is, there is not, as might be expected, a uniform wetting of the dry powder, but instead, the fine material is grouped in masses of varying size, and it is plain that some extraneous force would be required to break up these masses, since they evidently are in their present positions in obedience to the action of certain forces which have moved the particles from their original place.

The truth of this is easily demonstrable. Surface tension—that peculiar chain of action and interaction which causes rain drops to assume the spherical form—gathers the particles together in groups; and the finer they are, the more easily are they affected. The fineness of cement particles is, in this respect, a detriment to efficient hydration.

But in the process of this grouping, the particles have become slightly wetted. This has immediately resulted in reactions with the cement; and as the quantity of water is necessarily limited by the close grouping of these fine particles, a saturated solution between them, with deposition therefrom of the interlacing crystals before noted, has speedily resulted. In far shorter time, therefore, than is required for initial set under usual conditions, these little groups of particles have become consolidated; and unless the mechanical agitation of mixing is sufficient to break down this crystalline bond, they will remain unhydrated, save at their outer surfaces, so long as the concrete endures. This condition of affairs is difficult to overcome, even by the mechanical action of mixing, as the very small size of the groups renders unlikely their physical dispersion by the agitation of relatively large materials such as sand or stone, which by reason of the water present, are out of contact with one another. This is partially true regardless of the extent to which the process is prolonged. Further, the bond between these particles is speedily rendered more secure by secondary colloidal formations, with their desiccation by absorption of water by the interior of the mass. These colloidal boundary and bonding masses form the honeycomb structure seen in the photograph. Their appearance as white boundaries in some slides and dark boundaries in others is due to the method of polish-attack used in developing the surface prior to its examination.

This grouping and quick-cementing action, therefore, is the reason that the experimental apparatus developed proved inadequate for the task. This is particularly true of later forms, for in them the number of atomizing jets was increased to such an extent that the fine drops, though initially of a size suited to an individual cement particle, were in such numbers as to coalesce; and the larger drops thus formed caused, by surface tension, grouping of cement particles, such as has been shown. It is evident that means other than those at first proposed are required to produce effective hydration.

With a view to ascertaining the results produced by field conditions, as distinguished from the laboratory-made specimens which have thus far been examined, sections of a few concretes obtained from actual construction were examined. A large number of specimens of concretes were collected from representative structures throughout the world; and the author desires to take the opportunity here presented of extending personal thanks to the many engineers who have been so kind as to supply the samples asked for. Without their aid, the scope of this research would have been much restricted.

Examination of these specimens shows that similar conditions exist throughout the entire mass of each and since none of the specimens is over 1 cu. in. in volume, the quantity of cement remaining unhydrated through such grouping in each cu. yd. of concrete is a matter to ponder upon. It is easy to extend like illustrations to a far greater length with the data already at hand, but the series above given, covers a wide range of classes of concrete of varying ages

and conditions of service, and made under widely different conditions. *Grouping of particles in the manner indicated is a characteristic of all concretes so far examined;* and because of this grouping and of actions next to be indicated, it seems probable that only a small percentage of the cement added to concrete is effectively used.

Obviously, it is of the greatest importance that this grouping and isolation of large masses should be prevented. But even if this were quite perfectly done, there remain other actions which militate against thoroughly efficient hydration.

In microscopic examination of concretes, one of the most notable features of the hydrated matrix, aside from the large masses remaining unhydrated, is a "mealy" or speckled appearance. An example of this is shown at 60 diameters in Fig. 16 (concrete from pile caps underwater, Staten Island Ferry, Battery Park, New York). Such a surface is very difficult to photograph and was obtained only by the most delicate polishing. The mealy, or spotty, appearance is here clearly shown, and by careful examination, it can be determined that each of these spots is a cement particle, covered over with a colloidal, or perhaps a colloidal-and-crystalline skin, but remaining quite unhydrated at its center.

The reasons for these conditions are made clear by a consideration of the phenomena of hydration as before given, together with the absorption of water by the particle as hydration proceeds and the formation of an impermeable covering by the desiccation of this colloid through abstraction of water from the outer layers by the inner. Further, even with free access of water to the outside of a particle these conditions obtain, for in an agitated mass of cement in spite of the profuse formation of flocculent colloids, an unhydrated center has yet been found in each of the particles examined.

Therefore, it is evident that the finer the grinding, the better will be the hydration, as the finer the particle, the greater its surface in proportion to its bulk and in consequence, the more ready the penetration of reactive water. On the other hand, the finer the particles, the more ready and the closer will be their grouping by surface tension of the mixing water unless means can be found to lower this surface tension, without harmfully affecting the quality or the cost of the concrete.

The importance of increasing hydration cannot be exaggerated. Even a slight increase in thickness of hydrated layer results in notable increase in strength. In the earlier, though limited, investigations of this research when cement was delivered by an air-boil, to be met by finely atomized water under the best conditions for union, tensile and compressive strengths rose to high values, with a further peculiarity that the cement showed no retrogression with passage of time. Micro-sections also confirmed physical tests as to the progress of hydration, so that high hopes were entertained for commercial success along similar lines.

But since the conditions revealed by this research have become known and their causes have been ascertained, the next step is the overcoming of objectionable features by removing the causes of their origin. In other words, if high surface tension is at fault, to promote hydration lower that factor to a value which will not be disadvantageous. There are several substances which will lower the surface tension of water. One of these is alcohol, and the effect of additions of alcohol to the mixing water of cement may be readily investigated with the assistance of the microscope.

In Fig. 17 is shown the effect of adding water to this cement. As can readily be seen, the particles have grouped themselves in aggregations of varying size, in formations similar to those found in the examination of set concretes. But if to this water is added a small percentage of alcohol, dispersion immediately results, as in Fig. 18. Although the dispersion here is not complete, the cement is in vastly better situation to be effectively hydrated than it was when closely grouped, before the deflocculent (alcohol) was added.

In commercial work it is of course impracticable to use alcohol in sufficient quantities to properly promote hydration, nor would its use be beneficial to the concrete, because of the excessive liquid content thus made necessary to allow for the evaporation of the alcohol. But there are other agents even more efficient, which are cheap and easy to use and have no objectionable chemical action. So far as experiments have been carried out, these substances promise to produce concrete of greater density and strength than has heretofore been possible.

There are, however, certain other ordinary though important factors in the making of concrete which may be considered briefly. These are: *a* thorough mixing; *b* proper proportioning of aggregates; *c* proper consistency.

Thorough mixing is one of the most potent factors in securing thorough distribution of cement, thorough contact of cement paste with aggregate, and also, to a certain extent, hydration. Water does not necessarily wet aggregates instantly, for a reluctance or "skin effect" retards its passage over the surfaces of dissimilar materials. Further, this film of water on a piece of stone, for instance, or a particle of sand, holds beneath it air bubbles, which may in some instances be so numerous as to constitute an entire air-skin about the particle of aggregate, preventing contact between the cement and the sand or stone, unless mechanical agitation, with scraping by adjacent particles, is prolonged for a time sufficient to break down such films. Further, besides the smaller lumps and groups of cement particles caused by surface tension, there are commonly very large lumps of compacted cement in the sacks, and it often requires as much agitation to pulverize these lumps as it does to distribute the finer portions throughout the mix. A poor quality of concrete should not always be considered due to the cement, the aggregate or the water used, unless it is certain that proper mixing has been secured.

The proper selection and proportioning of aggregates is also of great importance. It has been pointed out above that void determinations were quite unreliable as a basis of proportioning, since the aggregates in the set concretes are all in far more dispersed positions, relative to one another, than they were when void tests were made. It is axiomatic that a concrete, which in density most nearly approaches the density of its large aggregate, is the strongest and best. Further, the smaller the amount of cement in a concrete, consistent with proper void-filling and proper bonding of aggregates, the better should be the concrete, since cement is the weakest of all the ingredients. For any given materials therefore the proportions should be determined in such ratio that maximum density will result, and if these proportions are adhered to in the field, a very superior concrete should be produced.

The question of consistency of mix is one which is closely related to the size and grading of aggregates. Obviously, a

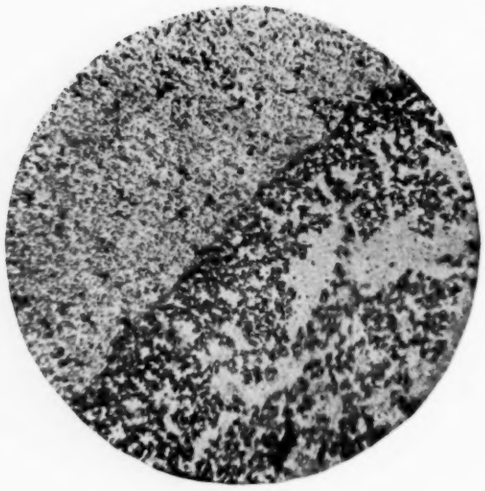

FIG. 15

FIG. 18

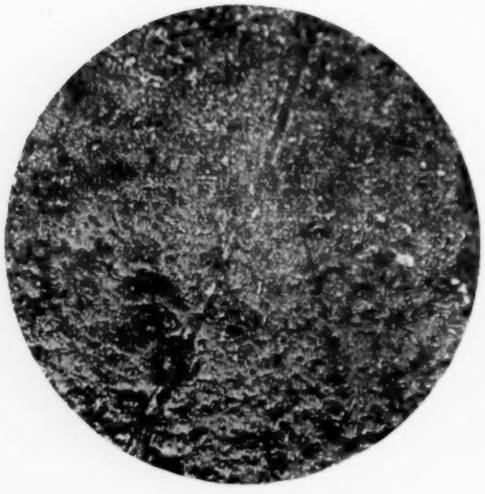

FIG. 16

FIG. 19

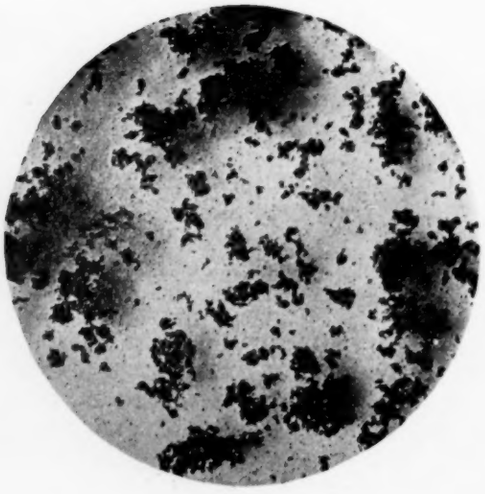

FIG. 17

FIG. 20

FIG. 21

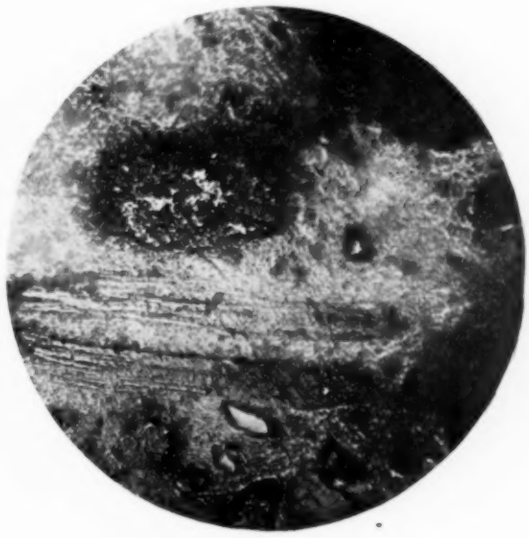

FIG. 24

FIG. 22

FIG. 25

FIG. 23

FIG. 26

fine sand, which has greater surface area than a coarse sand, will require more water for a given consisteney of mortar than would the latter. Similarly a graded sand having less voids will require less water than would sand in which all the particles are of one size, and the same remarks apply to the larger aggregate as well. Excessive water may further occasion shrinkage cracks in the matrix. Some of these may be so large as to be visible to the naked eye, but others whose existence is rarely suspected are renderd visible only by microscopic examination. Fig. 19 shows a three-branched shrinkage crack of this kind. If proper tests of materials are made, the correct proportion of water may be determined, as well as the proportions of the ingredients; and if proper supervision is maintained in the field so that these quantities are adhered to, there need be little fear of producing a concrete which will develop faults as it dries, of "drowning" the cement, or of failure properly to fill forms, or to bond with the steel. Eternal vigilance in these details is one of the requirements for success.

FIG. 27

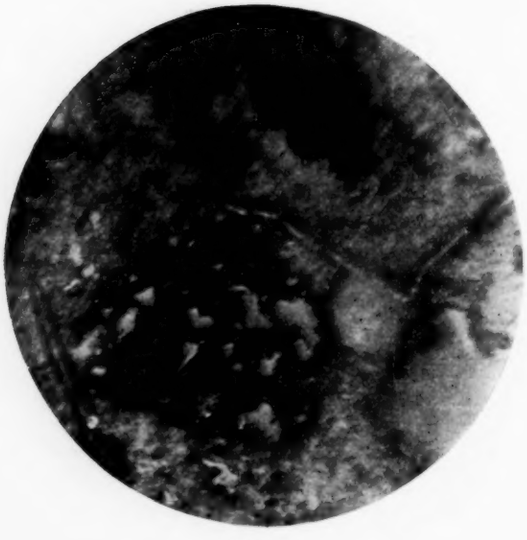

FIG. 28

There have been in the past certain cases of failure of concretes exposed to sea-water, which have become very widely known. It is probable that to no one phase of the concrete industry has so much thought and labor been devoted as to the study of the behavior of concretes in sea-water and the production of a cement capable of withstanding their action. There is considerable disagreement among authorities as to the causes of such disintegration, but there seems to be concurrence in the opinion that the formation of calcium sulpho-aluminate by interaction between the sulphates of sea-water and the aluminates of the cement, is in large measure responsible. This salt (calcium sulpho-aluminate) increases largely in bulk by crystallization and disrupts the concrete by the physical actions attendant thereon.

FIG. 29

If we were to picture mentally the formation of such an expansive material in the cement matrix of a set concrete, we should have to imagine a gradual straining of the material confining the crystals until rupture occurred. It is significant as to the correctness of such an hypothesis, that in the matrix of concretes which show outward signs of disintegration are found by microscopic study interior evidences of such strain.

In babbitts and the softer bronzes, incipient fracture is indicated by "shear planes" or "slip bands." In Fig. 20 at 150 diameters are shown similar "shear planes" in the cement matrix of a sea-water concrete taken from New York Harbor. It will be observed that these shear planes are at right angles to the polish-scratches on the specimen, so that their lack of identity cannot be questioned. Further, this is not an isolated case, since the same characteristic features are found in many other specimens taken from concretes in similar situations. Fig. 21 shows a repetition of these general features, as do also Figs. 22 and 23 all at 150 diameters. Other examples of the same structures could be readily furnished, but these four should be sufficient to establish the fact of their nature and existence.

Various stages of such crystalline disruptions, indicative at least of the possibility of such action, are detected by further micro-examination. In Fig. 24 at 200 diameters are seen a pair of parallel shear planes which have evidently progressed beyond the stage of strain and become actual fractures filled with crystalline material, as may be determined by their having withstood the abrasive action of the polishing powders, which cut away the softer matrix surrounding, leaving these fillings in high relief. In Fig. 25 at 200 diameters is shown a similar formation, with a large group of unhydrated cement particles near by. Fig. 26 shows at 150 diameters a similar conjunction of crystalline filling of slip planes and an unhydrated group. Whether or not this conjunction is of significance is a point yet to be fully determined.

Under certain conditions also this crystalline disruptive action may affect a considerable area. In Fig. 27 is shown at a magnification of 60 diameters part of such an extended action, where the crystalline formation is a long, disruptive dike. This structure resolved at higher magnification (200 diameters) is shown in Figs. 28 and 29, and this particular dike is of special interest in that it lies in a sample of a fresh-water concrete 12 years old, which is now showing pronounced outward signs of disintegration. It is greatly to be regretted that these dikes are so minute as to render their isolation and chemical analysis almost impossible. Such an analysis would be of great value in deciding many questions which at present seem to defy solution.

But whether or not calcium sulpho-aluminate is the major factor in disruptive formations in concrete, there seems to be agreement among cement experts that advantageous changes could be made in the chemical composition of cements, especially where they are to be used for sea-water concrete. A reduction in the alumina content would be beneficial in this respect, though it would entail serious difficulties in manufacture due to the higher temperatures required in burning the clinker. The addition in grinding of pozzolaine material, in order to obtain silica in a condition chemically available for union with the calcium hydrate liberated in the setting reactions, seems also to have much to recommend it. This latter change we should expect to prove very beneficial, for the leaching of calcium hydrate from cement structures which are subjected to the percolation of water is well known. Any such removal of material must necessarily leave a corresponding void suited to the reception of perhaps deleterious crystalline material, such as forms the dikes above noted.

There has recently been published a very exhaustive research, by Rankine and Shepherd[1] of the Geophysical Laboratory at Washington, on the products formed by the sintering of varying proportions of the three radicals, CaO, Al_2O_3 and SiO_2, which are the essential constituents of Portland cement. A portion of this three-component diagram, taken from the Rankine and Shepherd diagram and sketched with approximate correctness, is shown in Fig. 2.

In interpreting this diagram it should be noted that the lime content increases vertically, the percentage being read on the parallel horizontal lines. The percentage of alumina (Al_2O_3) increases downward at the right, the alumina content being read on the lines slanting downward from right to left. The silica content similarly increases downward but at the left, the percentage of silica for any mixture being read on the lines sloping from right to left. The heavy dotted lines indicate the loci of eutectic mixture, and the light dotted lines the approximate isotherms. It is evident from these that as the mixture under fusion changes its proportion of any of the three ingredients, the sintering temperature also changes; and, conversely, if the constitution of any mixture is known, the temperature required for its formation may also be determined.

For instance, the temperature commonly employed in burning Portland cement is in the neighborhood of 1500 deg. cent. This would be the proper sintering temperature of a mixture somewhere in the neighborhood of *D*, where the composition called for would be approximately 37.5 per cent of alumina, 6.5 per cent of silica and 56 per cent of lime. But the region of Portland cements is in the neighborhood of the point *A*, at which the indicated proportions would be alumina 9.66 per cent, silica 23.41 per cent and lime 66.93 per cent. For the proper formation of this latter compound a temperature of about 1900 deg. cent. would be required. As 1600 deg. cent. is the melting temperature of steel, and 1500 deg. cent. the usual kiln temperature, it is evident that in the burning of Portland cement, the natural tendency is to obtain a product unduly rich in alumina and low in silica and lime, so that the formation of the proper compounds is left largely to chance; and oftentimes unduly large proportions of Al_2O_3 are carried over, which, as was before noted, is generally held to be responsible for such disintegration of sea-water concretes as has been observed. This is a phase of the sea-water question in the solving of which the user can do little except demand a material of the quality he desires. Cement manufacturers are, however, caring for this question as adequately as commercial conditions will permit.

The above diagram, which is based on the tertiary alloy diagram of the late Dr. R. H. Thurston, is due to Prof. George B. Upton, of Sibley College, Cornell University. To him also is due the interesting transformation of chemical equations by reduction of certain elements to simpler equivalents, which does much to clear away the fog caused by the complex composition of cements and which further shows, by the same means, that the various cements, whether Portland, natural or iron-ore-Portland, have essentially the same actions as the basis of their setting properties. The location of these cements on the diagram are shown at *A* for standard Portland, *B* for the iron-ore-Portland used in the Panama Canal, and *C* for an American natural cement.

DISCUSSION

JOHN R. FREEMAN considered the paper, while excellently setting forth advanced methods in microscopic analysis of Portland cement mortar, gave simply a diagnosis of some of the most important diseases to which cement work is subject, without yet giving the remedy.

He looked upon the matter of applying the methods of micro-petography and micro-metallurgy to concrete as nothing less than epoch-making and considered that the recent studies made by the author and by certain other microscopists gave great promise of extremely important benefits. Yet so far as these studies were reported, he regarded them merely as an excellent beginning of a more precise, profound and intimate study of this most important building material, which should be encouraged by engineers and by cement manufacturers and carried very much further.

This paper considers chiefly the methods of microscopic

[1] Am. Journal of Science, Jan., 1914.

study of voids and lack of hydration. Another investigator has made a good beginning on studies of a soluble content of the cement, but it is no less important that the study should be continued into the realms of micro-physics and colloidal-chemistry, if one would learn the cause of some of the occasional serious failures in the strength of concrete and its slow disintegration by water and frost.

He had no doubt that ten years from now following studies of this kind, a better grade of concrete would be made and that some cement made by recent rapid and cheap methods and acceptable under present standard specifications would come to be regarded as doubtful if not dangerous, for portland cement is by no means a simple uniform material and different brands and perhaps different batches of the same brand are far from being alike and perhaps differ in very important particulars from the cement made ten, twenty or fifty years ago with which the reputation of Portland cement as a permanent water-resisting and frost-resisting material was established.

He recalled the great change that had come about within his experience of the past thirty or forty years in mixing and depositing Portland cement concrete; that the standard methods of distributing and depositing concrete in a sloppy condition which are standard today would not have been tolerated by any respectable engineer years ago in the days when standard practice was that of mixing concrete by shovels on a mortar bed, turning it over two or three times and depositing it in a mealy, slightly moist condition in thin layers and carefully ramming it until water flushed to its surface. Today the mixing by rotary machines surely is far more thorough and better than the old hand-mixing and this improvement doubtless more than compensates for the possible injury from excess of water in the modern method as compared with the old.

While nearly everyone knows of this great change in methods of mixing and depositing concrete, there are not many who fully appreciate the great change that has taken place within the past few years in methods of making cement in virtually a single operation under intense heat in rotary kilns, and we cannot yet be sure that the cement concrete put down today will all of it prove as durable as most of that made with the old-fashioned cement and deposited by the old-fashioned methods.

Microscopical studies like those presented in this paper will do much toward finding out how to improve the character of the cement and how to improve the quality of concrete, and today give more promise than any other line of investigation, but the studies must be carried much beyond the scope of those outlined in the present paper and the most refined processes of physical chemistry called in to aid.

The speaker suggested two matters of special importance which needed further study:

(1) That of the possible solubility of some of the ultimate compounds resulting from the setting of the modern cements, whereby structures having thin sections would in course of time lose much of the strength on which their integrity depends.

(2) The effect of obscure chemical properties in good-looking sand which might seriously affect the strength of the mortar.

To illustrate these matters he cited a few tests made, incidental to other work, in the laboratory of the New York Board of Water Supply and which suggested this danger of dissolving out and loss of the strength-giving quality in thin sections of concrete of somewhat pervious or lean quality. These tests were a few made primarily to determine whether or not aggregates containing certain varieties of limestone were soluble, but in the series were specimens made up for control purposes with aggregates essentially insoluble. Croton water under about 40 pounds pressure was made to percolate very slowly through disks some 4 or 5 in. in diameter by 10 or 12 in. broad for a number of months and the filtrate caught and subjected to chemical analysis, with the result that it appeared that more of the cement than of the aggregate was being dissolved. At the close of this test a few of the disks were tested for compressive strength with the surprising result that they possessed, as the speaker remembered it, less than one-fourth of the strength of similar disks which had not been subjected to this slow percolation. These experiments were not followed up because of lack of time and pressure of other work, but they give much food for thought, and it would seem that, after all this percolation, the cement in these specimens must have been hydrated and thus free from one important defect found by the author in most of his specimens.

The speaker recently has been shown the results of an investigation of disintegration of concrete which seemed to clearly indicate that the cement was partially dissolved out near the surface after a few years exposure to water and that the action of frost on the slightly porous mass remaining caused its breaking down.

Upon the second topic, of obscure chemical properties, and as illustrating the need of supplementing the microscopic studies by some researches in colloidal chemistry or by researches advanced into the realms of physical chemistry, he cited some troubles that he encountered three or four years ago when planning a concrete dam. Close to the site he found what seemed to be a perfect mixture of gravel and sand, well graded and apparently clean and requiring simply to be mixed with cement and water in order to give excellent concrete, but on sending samples of this material to the laboratory for mechanical analysis and test of mortar it was found that the mortar made from this sand would not stand up sufficiently to be held in the clamps of a testing machine after setting 7 days, and after setting for 28 days it gave a strength less than one-fifth of the normal when tested with four or five different brands of cement; yet one particular brand of cement was found which gave a fairly strong mortar with this same sand, thus proving that the trouble was to be looked for in the sand rather than in the cement, but no reason has yet been found why this one particular make of cement worked so very differently with this peculiar sand.

After washing this sand in a way intended to cause mutual attrition and scrubbing of the individual grains, it was found to give tolerably strong mortar with all these brands of cement, and on analyzing the wash water by methods used for drinking water the reports showed a large percentage of albuminoid ammonia. An expert of the U. S. Department of Agriculture, who had spent much time on studies of the solubility of the mineral ingredients of different soils, suggested to the speaker that the peculiar behavior of this sand in relation to Portland cement was probably due to a low, complex organic acid, akin to tannic acid, which was present in a condition of *adsorption* on the outside of the sand grains and that to find out what was

worth knowing about this action might keep a skilled research chemist busy for a year.[1]

It is certain that some concrete has withstood frost and the wear of rapid currents of water for many years marvelously well. On the other hand, it is certain that some concrete from cement made by rapid modern methods, under rigid specifications and conscientious inspections, has gone to pieces in a most distressing way. It is possible that some of the modern concrete put down with a super-abundance of water, and perhaps some that is made with cement containing ingredients leading to compounds that will ultimately become soluble, may bring disaster when concrete is used in thin sections under high water pressure and expected to retain its normal strength indefinitely.

The subject is extremely important and engineers should be grateful for all contributions to it like the present paper and should be pleased to note that while the author now simply gives his method of diagnosis, he implies that in a future paper a remedy for some of the present troubles will be presented.

FRANK B. GILBRETH said that his first experience with Portland cement was in 1885, when cement practice was entirely different (from a workman's standpoint) from what it is to-day. In those days any man that used Portland cement on the first set was considered an ignoramous; to-day any many that uses it on the second set is considered a fool.

This work on the chemistry of cement is of tremendous importance to the country. What we are going to do with cement in the next few years as compared with what we have done will be of tremendous value in the structural world.

He had specialized in concrete, not in the chemistry of it but in its utilization, and in the latter connection he gave some of his experiences.

He was once building a hotel in San Francisco, and the reinforced concrete columns and girders were built up to the second floor, and the rest of the building was proceeded with, but the concrete did not set—and had not done so for eight weeks. Forms were taken out in a few places to see whether air was required to dry it out, but without effect. Just as it seemed that the concrete work must all be taken down, the San Francisco earthquake occurred, and the day after the concrete was found to be set. Whether it happened that the earthquake caused the setting or no, the fact is exactly as stated.

We are not sure but the atoms that make up the molecules in concrete might possibly require a physical start to cause the material to set. Several times the setting of concrete that has been delayed has been known to be started by machinery being run in the vicinity, and it is well known that if air-riveters are working on the forms of concrete when the latter is being poured, a quicker set is obtained than at another time.

F. H. NEWELL also related an experience in connection with concrete refusing to set. The Board of Engineers had set up some concrete work at a large expense to the Government, and then reported that the concrete did not set and must be taken out. By the time the report was issued, the piers which were officially condemned as worthless had set as hard as granite.

In reference to the fineness of the cement, he thought it is generally recognized that fine cement is so much better than coarse cement such as that which stays at the top of a 200 mesh sieve, and which has very little cementing value. Such cement ground up and passed through a 200 mesh sieve and made into little test pieces had been found to break at 600 lb. per sq. in.; reground, passed through the 200 mesh sieve and set up again, test pieces broke at perhaps 300 lb. per sq. in. Reground a third time and set up, pieces broke at about 200 lb. This illustrates the point that the regrinding of the cement, bringing the minute particles into immediate contact, enables it to set up again, and apparently indicates that we are not getting the full cementing value from our concrete as ordinarily used. He thought this point is one worthy of a good deal of investigation.

G. A. RANKIN[1] (written): This paper is very interesting and instructive in so far as it gives a large amount of data on the mechanical hydration of Portland cement and the disintegration of concrete in sea water. The discussion of the components of cement CaO, Al_2O_3, SiO_2, F_2O_3 and MgO, is, however, open to criticism.

The statement that if the constitution of any mixture is known, the temperature required for its formation may also be determined, is not true. The temperatures given in Fig. 2 are temperatures of complete melting and not temperatures required for the formation of compounds. In fact, it cannot be said that there are any definite temperatures for the formation of the compounds which go to make up Portland cement, as the formation depends on the time of heating and the fineness of the raw material, as well as the temperature. If sufficient time is allowed, it is possible to form these compounds even at temperatures much below that at which melting begins in a given texture. In actual cement practice, however, the temperature of burning is sufficiently high so that a small amount of the clinker is melted. This is necessary, as a small amount of melted material acting as a flux is required in order that the formation of compounds may be completed within a reasonable time. The temperature of the clinkering zone as determined by the Geophysical Laboratory in a number of Portland cement kilns was found to be about 1425 deg. cent. At this temperature only a small portion of the clinker is fused (melted); even so, it is sufficient for the proper formation of the compounds if the raw material is of a fineness such as is ordinarily required in good cement practice; in other words, the reactions which take place in the burning of Portland cement clinker go practically to completion even at a temperature of about 1425 deg. cent. The statement that the formation of the proper compounds is left largely to chance and that oftentimes unduly large proportions of Al_2O_3 are carried over is not true of good cement practice. In such practice it is possible to so control the raw material and conditions of burning that the resulting cement will be made up of proper compounds in such proportions that no improperly combined Al_2O_3 will be present. As to whether or not it is the compounds of Al_2O_3 which react with sea-water to cause the disruption of concrete is another matter, which it should be possible to settle by determining the action of sea-water on cements made up of the two aluminates, 3CaO, Al_2O_3, and 5CaO, $3Al_2O_3$, which occur in Portland cement. The idea that CaO and MgO and likewise Al_2O_3 and Fe_2O_3 act similarly

[1] See Eng. News, July, 1912, contribution by J. R. F.

[1] Geophysical Laboratory, Washington, D. C.

in hydraulic cements and form similar chemical compounds is erroneous. Recent investigations have shown that MgO and CaO, likewise Al_2O_3 and Fe_2O_3 do not act chemically alike, either in the reactions which take place in the burning of cement or in those which take place during hydration. For example, CaO unites with Al_2O_3 in the formation of four distinct compounds, while MgO unites with Al_2O_3 to form but a single compound whose properties are not similar to the calcium aluminates. Again, Fe_2O_3 and Al_2O_3, as they occur in cement compounds, react with water very differently, the ferrates being slow to act while the aluminates react with great rapidity, and the final products of hydration have decidedly different properties. It will be seen, therefore, that speculation which assumes the reactions between the chemical components of Portland cement, based on any two or more of these components acting as chemical equivalents, may lead to disastrous results.

In order to establish a sound basis for the study of Portland cement and its use in concrete, it is absolutely necessary that one proceed systematically to determine first the components which make up the cement. These are CaO, Al_2O_3, SiO_2, Mgo, Fe_2O_3, etc.

Knowing the components, the next step is to determine by experiment the compounds formed by these components in the proportions as they occur in cement. Having isolated each of these compounds to determine its composition and its optical properties for its certain identification, it will then be necessary to study its hydraulic properties. With this data it will be possible to state definitely just what is the nature of the setting of cements, also to state just what compounds should be present in order to produce the strongest cement and what compounds are best to withstand action of seawater, etc.

Proceeding in this systematic way, work has been carried on at the Geophysical Laboratory in an effort to determine the compounds formed by the components which make up Portland cement clinker. Since the components CaO, Al_2O_3, SiO_2 make up over 90 per cent of the chemical composition of most Portland cement, it seemed best to first make a study of these three oxides by themselves. The result of this investigation has shown that when CaO, Al_2O_3, SiO_2, are burned in the proportions as they occur in Portland cement clinker, the resulting product is made up largely of the compounds $3CaO.SiO_2$; $2CaO.SiO_2$; $3CaO.Al_2O_3$, with occasionally small amounts of free CaO and the compound $CaO3Al_2O_3$. Each of these compounds has definite optical properties which have been determined; this enables one to identify these compounds in any mixture in which they may occur.

The cement clinker made up only of CaO, Al_2O_3, SiO_2 on grinding and mixing with water has been found to possess all the properties of a desirable Portland cement. It has furthermore been found by subsequent work of the Bureau of Standards at Pittsburgh, that over 90 per cent of an average Portland cement clinker is made up of $3CaO.SiO_2$; $2CaO.SiO_2$; $3CaO.Al_2O_3$; and $5CaO.3Al_2O_3$. These compounds are present as definite individual crystals whose optical characteristics are not appreciably affected by the small amount of MgO, Fe_2O_3, etc., which are always present in Portland cement. A study of the hydration of those compounds was undertaken by the Pittsburgh Laboratory of the Bureau of Standards, and from the results obtained it is now possible to explain the chemical reactions which take place during the setting of Portland cement. With these added data it is now possible to proceed with confidence to the determination of the proportions of these CaO, Al_2O_3, SiO_2 compounds which should be present in order to produce a cement which on setting will possess the most desirable properties for various purposes.

HARRY FRANKLIN PORTER[1] wrote that Mr. Johnson has established, by laboratory methods, conclusions which the writer believed since 1908, that the true cement is a colloid produced by thorough hydration. A special article dealing with this important subject, based largely on the findings of Dr. Michaelis, was inserted in an appendix to a publication by the writer the following year.

Further, at the 1910 convention of the American Concrete Institute, the writer again devoted a considerable portion of a paper to a discussion of this subject, recounting an experience in Philadelphia several years previous, in which the value of thorough mixing of concrete as promoting colloidal formation was conclusively demonstrated.

This experience was that a certain batch of concrete, supposedly useless on account of having been churned in the mixer during the whole of the noon hour, was dumped on concrete poured the previous day. Instead of becoming a loose mass of rock and sand, as expected, it set up smooth and hard and adhered so firmly to the old concrete as to require picking to remove it. Moreover, it presented a different appearance than any concrete the writer had seen before and was much more uniform in color and texture. The exceptionally thorough mixing, and the unintentional thorough hydration, instead of weakening it seemed to have vastly improved its quality.

In concrete work, there is necessity for, first, intelligent selection of the aggregates; second, their scientific proportioning; and third, their thorough admixture. If the proper care is taken in these matters, not only is a considerably less amount of cement sufficient, but a concrete of vastly superior quality and strength is secured.

THE AUTHOR. The comment most frequently met with in regard to this microscopic study of concretes is that it does not indicate a remedy for the defects shown. Microscopic examination is primarily a diagnosis. It shows concretes as they actually are, not as they are supposed to be; and the faults that careless procedure and improper selection of materials induce. Further, the inter-relation and the inter-action of the several materials composing concrete are made plain, together with the penalties attendant upon any neglect of these inter-relations.

The remedy lies very largely in alterations of our present procedure. It is very true that there are certain beneficial changes that might be made in the cement, which is not by any means an ideal product, but it is true, also, that more care should be used in the selection and particularly in the quantities of the other constituents used. When the concrete maker realizes that the quantities and gradings of his stone and especially of his sand are vitally important and when he also realizes that the quantity of water may not be varied beyond certain limits, we may hope to see a very important change in the quality of concrete.

It should also be realized by engineers that concrete is not an infinitely dense, hard, resistant substance, but that, on the contrary, it is a material which varies in durability.

[1] *Industrial Engineer*, Chicago, Ill.

DESIGN OF RECTANGULAR CONCRETE BEAMS

BY HOWARD HARDING, ROCHESTER, N. Y.

Associate-Member

THE resisting moment of a reinforced concrete beam, in inch-pounds, may be represented by the formula

$$\text{Resisting moment} = R\ b\ d^2$$

where

b is the breadth of the beam in inches
d is the effective depth in inches
R is a numerical coefficient

The value of R (for a given ratio of the modulus of elasticity of steel to that of concrete) depends upon the percentage of steel reinforcement used and the safe working stresses for steel and concrete. These values of R for different working stresses and percentages of steel have been plotted in very

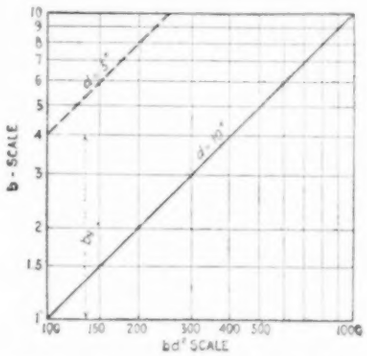

FIG. 1 DIAGRAM FOR CHECKING RELATIONS INDICATED BY EQUATION

convenient curve form in Turneaure and Maurer's "Principles of Reinforced Concrete."

For the purposes of this article it will be necessary to cite only a few values of R corresponding to some of the more common working stresses. The ratio of E_s to E_c is taken as 15.

For $f_c = 700$ and $f_s = 16{,}000$
For $f_c = 600$ and $f_s = 16{,}000$
For $f_c = 500$ and $f_s = 16{,}000$
For $f_c = 400$ and $f_s = 16{,}000$

$R = 120$, per cent steel $= 0.87$
$R = 94$, per cent steel $= 0.68$
$R = 71$, per cent steel $= 0.50$
$R = 49$, per cent steel $= 0.34$

In case of reinforcement in excess of the amount required to divide the working stresses as shown there is a slight gain in strength due to the shifting of the neutral axis toward the steel. When the concrete has reached its working stress the steel will still be understressed. In such cases the value of R used must be the one corresponding to the working stress of concrete and the percentage of reinforcement.

The ordinary method of using the data thus given is that of "cut and try."

After quite a bit of experience with that method the growing dissatisfaction prompted the plotting of the equations involved so as to give a direct method of arriving at the result desired. After some study a logarithmic form was adopted which seems to serve the purpose.

On a sheet of logarithmic cross-section paper the values of b were denoted by the horizontal lines intersecting the left-hand ordinate. The left-hand ordinate was called the b scale. Through the origin (that is, through the point 1, 1 of the diagram) draw a line to the right at 45 deg. to the horizontal. By inspection of Fig. 1 it can readily be seen that the point of intersection of any value b with the oblique line, if projected vertically upward to the top horizontal scale of the paper, will give a logarithmic reading exactly equivalent to that of the b scale. Now let the oblique line correspond to $d = 10$ and multiply the readings of the top horizontal scale by 100. Then again by inspection, it will be seen that by projecting vertically upward the intersection of any value b and the line $d = 10$, we may (on the multiplied horizontal scale) read directly the corresponding value of $b\ d^2$.

For values of d other than 10 there is a family of lines parallel to the line $d = 10$.

It now remains to introduce the factor R into the diagram. A range of from 40 to 120 for this coefficient will be found to cover about all cases likely to arise. Since the bending moment is ordinarily computed in foot-pounds we may write our fundamental formula to correspond to those units. Then

$$\text{Resisting moment} = R\frac{b\ d^2}{12}$$

and when $R = 120$

$$\text{Resisting moment} = 10\ b\ d^2$$

By projecting the values of the $b\ d^2$ scale upward to a horizontal scale whose readings are 10 times that of the $b\ d^2$ scale, we obtain directly the resisting moment in foot-pounds corresponding to any value of $b\ d^2$ and $R = 120$. The diagram is arranged to care for other values of R by taking the proper fractional part of the resisting moment of the beam at $R = 120$. Thus, for $R = 60$, one half of the resisting moment of the beam at $R = 120$ is taken. This is done graphically by proceeding upward to the horizontal line representing the desired value of R and thence proceeding along the oblique line to the top scale where the resisting moment in foot-pounds is read.

In the lower part of the diagram it will be noticed that there are two other families of oblique lines. One of these shows the ratio of d to b and the other the cross-sectional area or the product of d times b.

The method of using the diagram is ordinarily just the reverse of the order in which it has been developed. First decide upon the working stresses and determine the corresponding value of R and the percentage of reinforcement. Then compute the bending moment in foot-pounds which it is desired that the beam should withstand. Locate the bending moment thus computed at the top of the diagram, Fig. 2. Thence proceed parallel to the oblique lines until the value of R previously determined is intersected. From this intersection proceed vertically downward into the lower part of the diagram intersecting the horizontal lines representing breadth, and the oblique lines representing depth. At any one of these intersections the required breadth and depth may be read. Any of the corresponding values will satisfy the condition for strength, but the values chosen will depend upon the shape of beam desired. If we wish a certain

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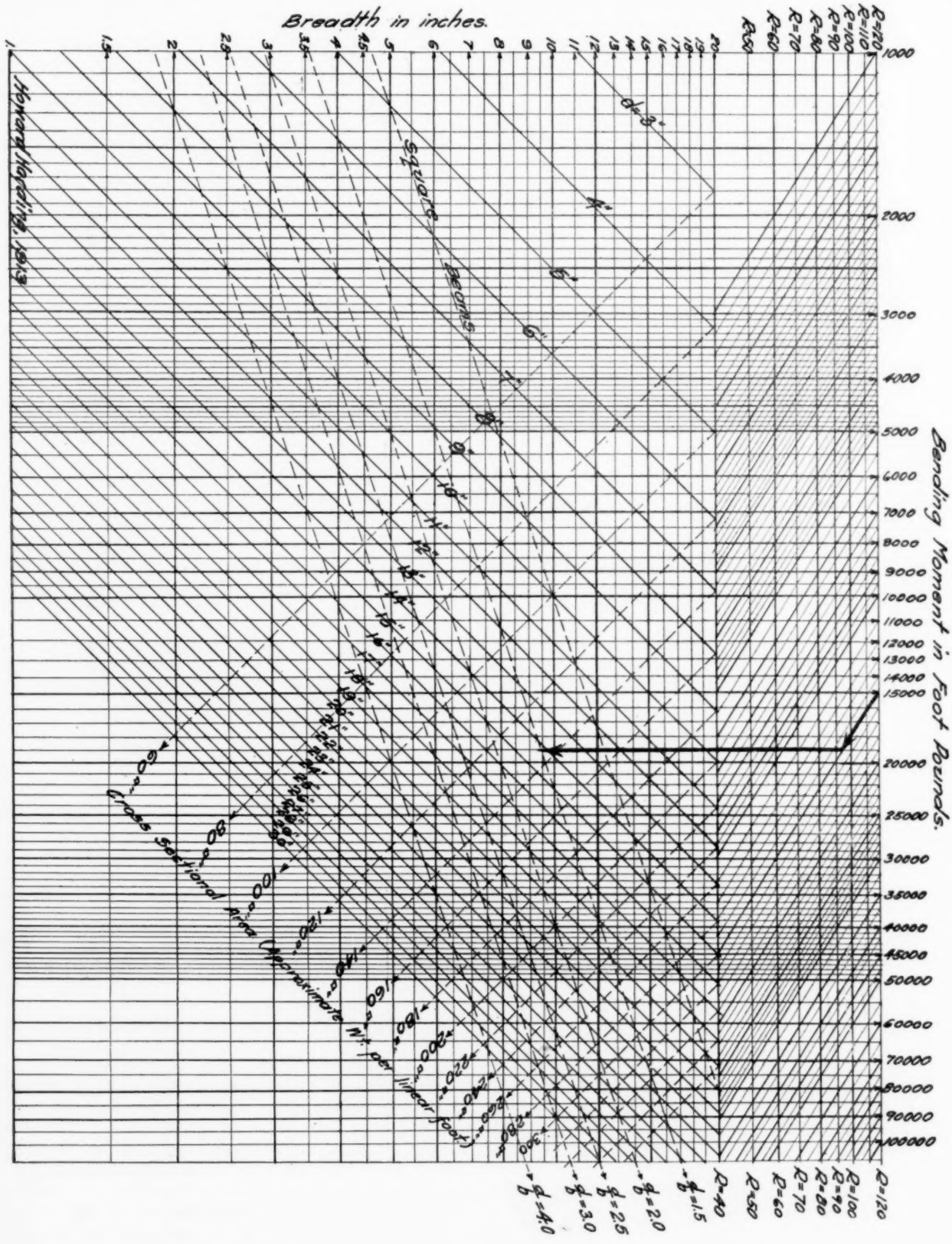

FIG. 2 DIAGRAM FOR GRAPHICALLY DETERMINING THE RESISTING MOMENTS OF REINFORCED CONCRETE BEAMS

ratio of depth to breadth we can obtain it by stopping at the dotted line representing that ratio. If a certain depth or a certain breadth is desired it is necessary to proceed vertically downward until said depth or breadth is intersected and the point of intersection automatically determines the value of the other dimension.

In a beam of dimensions as determined by the use of the diagram in Fig. 2 no allowance has been made for the bending moment of the beam due to its own dead weight. To take care of this additional bending moment either one or both of the dimensions must be increased. To determine the amount of the increase necessary for different spans and types of beams a second diagram has been developed. The strength of the beam may be increased in three different ways, viz; (*a*) by increasing both breadth and depth, (*b*) by increasing depth only, and (*c*) by increasing breadth only. The determination of multiplying factors for these three cases is now taken up.

The author derives the following formulae to cover these cases:

Increasing strength by increasing breadth and depth:

$$\left\{\frac{(1.56\ k^3)}{k^3-1}\right\} l^2 = R\ d$$

Increasing strength by increasing depth only:

$$\left\{\frac{(1.56\ k)}{k^2-1}\right\} l^2 = R\ d$$

Increasing strength by increasing breadth only:

$$\left\{\frac{(1.56\ k)}{k-1}\right\} l^2 = R\ d$$

It will be noted that the form of the three expressions just developed is the same, the only variation being in the exponent of the term k enclosed in the parenthesis. A tabulation of the values of the parenthesis corresponding to different values of k is given in the paper.

To summarize the foregoing it may be well to work out a problem in detail. The writer has found the following to be a very convenient and useful form of computation:

Girder G_1

Span = 18 ft. Load = 370 lb. per linear ft.

Bending moment $= \frac{w\,l^2}{8} = \frac{370\times 18^2}{8} = 15{,}000$ ft-lb.

Shear = 3,330 lb. + ½ weight of beam

$f_s = 16{,}000$ $f_c = 600$ $R = 94\%$ steel = 0.68 sq. steel rods

$b_1 \times d_1$	12.5×12.5	9.4×14.10	7.5×16
Corrector	1.17	1.145	1.136
$b \times d$	14.6×14.6	10.8×16.1	8.5×18.2
$b \times d$	213	133	120
Unit shear	25	35	37
Steel	4⅝-in. bars, or 3 11/16-in. bars	3 9/16-in. bars, or 2 11/16-in. bars	3 9/16-in. bars, or 2 11/16-in. bars

The above tabulation shows three beams which would be suitable. The one which best suits the rest of the structural design may be used. The correction factor used was the one corresponding to the increase in both dimensions. It should be noted that d represents the effective depth, and the distance from the steel reinforcing to the bottom of the beam must be added to get the total depth. This "cover" portion of the beam has been neglected in all of the computations for strength. The determination of the second beam in the tabulated form is plainly indicated on the diagrams so that the method of using them ought to be readily understood.

MODEL EXPERIMENTS AND THE FORMS OF EMPIRICAL EQUATIONS

BY EDGAR BUCKINGHAM, WASHINGTON, D. C.

Member of the Society

THE purpose of the paper is to illustrate, by application to a few problems in technical physics, the methods by which dimensional reasoning may be made use of in planning and interpreting experiments, in devising empirical equations for use in design, and especially in the performance of experiments on models in such a way as to furnish reliable information about the full-sized originals.

The General Theorem. The well known principle of the dimensional homogeneity of physical equations may be expressed in the form of an algebraic theorem which is convenient for application. This will be referred to as "the Π-theorem": it states that any relation which involves n physical quantities of different kinds, may be described by an equation of the form

$$f(\Pi_1, \Pi_2, \ldots \Pi_{n-k}) = 0,$$

in which k is the number of fundamental units needed, and the Π's are dimensionless products of powers of the symbols which represent the various physical quantities involved in the relation. The value of k is generally 3 in mechanical problems, though in statics $k = 2$. In the most general problems, involving thermal and electromagnetic, as well as mechanical quantities, $k = 5$.

To avoid delay at this point, the deduction of the Π theorem is relegated to an appendix.

The Flow of Liquids in Smooth Pipes. The mechanism of applying the Π-theorem in a concrete instance is illustrated in this familiar example, and it is shown that any correct equation for representing the flow of an incompressible fluid through a smooth pipe must be of the form

$$\frac{DG}{\rho S^2} = \phi\,\frac{DS\rho}{\mu}$$

in which: D = diameter; G = pressure gradient; S = speed of flow; ρ = density of the liquid; μ = viscosity of the liquid; and ϕ represents an unknown function of $\frac{DS\rho}{\mu}$ which must be determined empirically. Comparison with the best experimental data confirms the conclusion, and the plotting of a diagram, in which English and metric data are used indiscriminately and without conversion, illustrates the convenience of working with dimensionless quantities. The comparison of the above equation with the known facts helps us to a physical understanding of the phenomena of stream line and turbulent flow—especially the existence of a critical speed and the unimportance of temperature at high speeds or in large pipes.

Resistance of Immersed Bodies at Moderate Speeds. A "moderate" speed is one far enough below that of sound in the medium, that the compressibility of the medium has no sensible effect on the resistance encountered by a moving body. The Π-theorem is used in finding the necessary general form of empirical equations between the resistance, speed, size, and shape of the body, and the properties of the

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medium. Consideration of the general equation and of the experimental work needed for determining its specific form, leads to the notion of model experiments and to the conceptions of corresponding speeds and of dynamical similarity. This explains the justification of the method of determining data on air-ship forms by the use of very small scale models in water.

Resistance to the Flight of Projectiles. This section shows how the conclusions reached in the preceding one must be supplemented and modified when the speed approaches or surpasses that of sound and the resistance is increased by the continual drain of energy in maintaining the head and base waves. Model experiments in air are impracticable, but the interesting possibility suggests itself that information about the resistance of large projectiles might be obtained from observations on the retardation of very small models—to a scale of about $\frac{1}{60}$—shot under water. A plot of observed resistances in air shows that the resistance offered to high-speed projectiles depends on density and compressibility while the effect of viscosity is negligible.

Screw Propellers. The consideration of the thrust of screw propellers illustrates an application of the Π-theorem in a more advanced case, the number of kinds of quantities to be considered being greater than in the examples already treated. The conditions for dynamical similarity of propellers of the same shape but different diameters are found, the simplest being that the propellers shall be run at the same slip ratio and the same relative immersion. The conditions can not all be exactly satisfied in practice, but it is further shown how one of the conditions, that relating to viscosity, is of small importance, and how by neglecting it, approximate results may be obtained. The condition for corresponding speeds of advance is then the same as in Froude's law of comparison, and the thrusts are proportional to the cubes of the diameters. This conclusion appears to agree with experiment, showing that the approximation made in neglecting viscosity is permissible. If propellers of a given shape are run at the same slip ratio and so deeply immersed as to cause no surface disturbance, all speeds correspond, and the thrust is proportional to $(DS)^2$, D being the diameter and S the speed of advance.

Heat Transmission. The Π-theorem is applied to a simplified problem in heat transmission, and it is shown that while various roughly approximate relations must be assumed, the assumptions are plausible and the results obtained probably nearly correct for ordinary heat transmitting devices, such as steam coils with air forced over them for indirect heating, under working conditions. Some of the algebraic expedients which are occasionally needed in using the Π-theorem and which have not come up in any of the preceding sections, are illustrated here. The physical conclusion is that under ordinary working conditions, when a fluid is forced through a heating or cooling device of given design, the difference of temperature of the fluid and the solid surfaces at exit is a definite fraction of the difference at entrance, this fraction being fixed by the design of the apparatus and influenced in only a very minor way by the size of the apparatus, the initial difference of temperature, the nature of the fluid, or its rate of flow.

Conclusion. It is pointed out that the method of reasoning by means of dimensions is purely algebraic and therefore rigorous, so that the correctness of conclusion obtained by its aid depends only on how much physical knowledge and common sense we display in using it, the method itself being a mere logical tool. It is, however, extremely easy of application, and it often throws a quite surprising light on experimental facts which would, without its aid, be very difficult to analyze and interpret correctly.

DISCUSSION

M. D. HERSEY. The author has struck the keynote of a new development of technical physics, which will eventually play the same part in mechanical engineering that physical chemistry has begun to play in the chemical industries. It will directly be seen that this prediction is not a prophecy at all, but merely an observation.

The importance of technical physics, as a branch of subject matter, is already so clearly recognized in Germany that laboratories are being established exclusively devoted to this field.

But the development which I think we may now anticipate is something distinct from this, and a natural sequel to it: I refer to the development of technical physics, not as a branch of subject matter, but as a method of reasoning.

It is from such a standpoint that technical physics becomes analogous to physical chemistry which is the planning and interpretation of chemical experiments in the light of thermodynamics and the phase rule; and the Π-theorem is closely analogous to thermodynamics and the phase rule. Thermodynamics affords certain rigid connecting links between seemingly isolated experimental results, while the phase rule tells us the number of degrees of freedom of a chemical system. The Π-theorem likewise affords rigid connecting links which not only serve as a check on the consistency of our results, but may greatly cut down the labor of experimenting. Thus, on applying the Π-theorem to lubrication, we find, under certain conditions, that the coefficient of friction, f, must be some function or other of the *two* variables $\frac{\mu n}{p}$ and $\frac{D^3 n}{Q}$ alone; in which μ denotes viscosity, n revolutions per unit time, p bearing pressure, D journal diameter, and Q volume of oil pumped through bearing in unit time. Hence, the same change in f will be produced by a given change in the argument $\frac{\mu n}{p}$, whether this change is, in turn, caused by varying μ in one direction or by varying p in the other direction; and so on. These facts, all implicitly contained in the Π-theorem, can, for the sake of emphasizing our analogy, be expressed by the equations

$$\frac{\delta f}{\delta \mu/\mu} = -\frac{\delta f}{\delta p/p} \quad \text{whence} \quad \frac{\delta f}{\delta p} = -\frac{\mu}{p}\cdot\frac{\delta f}{\delta \mu};$$

$$\frac{\delta f}{\delta D^3/D^3} = -\frac{\delta f}{\delta Q/Q} \quad \text{whence} \quad \frac{\delta f}{\delta D} = -3\frac{Q}{D}\frac{\delta f}{\delta Q}; \text{ etc.}$$

And, just as the phase rule tells us that the number of degrees of freedom in a chemical system is $F = K - P + 2$; K being the number of components which coexist in phases; so also the Π-theorem tells us that the number of degrees of freedom in a physical system is $f = p - k - 1$; k being the number of fundamental units needed to describe a relation subsisting among the p physical quantities. The number of factors which must be experimentally varied in order to map out the relation for the first time is, of course, one

greater than the number of degrees of freedom, or simply $p - k$.

The author has himself stated that the paper contains nothing essentially new. Any allusion to the contrary would be an impediment to the successful use of the methods presented. The kernel of the paper is a theorem which is merely a restatement of the requirements of dimensional homogeneity, announced by Fourier nearly a hundred years ago, and extensively used by Rayleigh and others. But the fact that the paper contains nothing essentially new does not diminish its value. Gibbs' phase rule, too, was new only in form, not in substance, yet it served as the crystallizing influence which caused an immense number of latent ideas to fall into line, and we may expect the Π-theorem to play a similar role.

This inevitable development of technical physics into a unified branch of science, which will require the same fundamental place in the engineering curriculum that physical chemistry now holds in the chemical curriculum, can be facilitated if writers on the problems of hydro and aero dynamics, heat transmission and the like will be as introspective as possible, explicitly calling attention not only to their results, but to their methods of reasoning as well. For in every successful artifice of reasoning there must be some element which is universal, and capable of being generalized and made into a working tool.

In conclusion, the fact should be recorded that the writer's paper on the Laws of Lubrication is in part an application of the Π-theorem, and that, therefore, the discussion of his paper is to be considered a continuation of the discussion of Dr. Buckingham's paper.

MELACH I. NUSIM. The author calls attention to the aid to be obtained in the application of the theory of dimensions in scrutinizing results of tests, and his general methods should be of value to engineers engaged in experimental work.

The conditions for similarity, discussed in the paper, have been noted and applied with great advantage by designers of centrifugal compressors and pumps. Two centrifugal compressors, if they are geometrically similar, have the same efficiency provided the following relation is maintained between the rated flow, Q, the peripheral speed of the impeller, S, and the impeller diameter, D:

$$\frac{Q_1}{S_1 D_1^2} = \frac{Q_2}{S_2 D_2^2} = \text{constant.}$$

The experimental data on one particular size of compressor can be utilized to predict with accuracy the performance of a number of sets provided they are made geometrically similar and rated according to the relation mentioned above. In terms of the mean effective pressure, P, the r.p.m., N, and the flow, Q (volume per unit of time), the relation is equivalent to

$$\frac{QN^2}{P^{\frac{3}{2}}} = \text{constant.}$$

A. R. DODGE said he had investigated the drop in pressure of superheated steam under similar conditions to those of Stanton and Pannell's experiments on air and water, using a 1-in. smooth drawn brass pipe, steam jacketed. The results obtained showed that the same formula for pressure drop would apply for steam, in addition to air, water and oil. It ought, therefore, to apply for any fluid and simplify existing formulae.

He had also made a number of steam tests with commercial iron pipe, 2 to 8 in. in diameter, and found that the results were consistent, being only modified by the roughness of the pipe.

JOHN R. FREEMAN said that he possessed data of his own experiments with ordinary rough pipes, which might be of value in solving the problems presented. He said he had conducted a very extended range of experiments, in 1893, on pipes of all degrees of smoothness, from seamless brass pipe to exceedingly rough pipe.

L. W. WALLACE asked for further information in connection with investigations to determine certain facts in reference to locomotive sparks. It becomes necessary to know to what height sparks are ejected from locomotives under various operating conditions, and he asked whether experiments with a specially designed model locomotive would give data that would be comparable with the actual height the sparks would be thrown, the size of the sparks, etc.

THE AUTHOR. In reply to Mr. Wallace, it is conceivable that model experiments might be so devised as to furnish the desired information; the difficulties appear, at first sight, rather formidable. It is impossible to say off-hand, before examining the problem carefully, whether an attempt to solve it in this way would have any prospect of success. It would seem much simpler to study the actual emission of sparks from a locomotive by making runs at night and taking photographs—possibly kinematograph records—from two points on the train, one close to the locomotive and one much farther back.

In reply to Mr. Freeman, in trying to get an equation which could be made to represent the resistance of both smooth and rough pipes by varying only a single quantity—representing the roughness—the author had used Saph and Schoder's results exclusively, because in a preliminary study consistency was more important than accuracy. He had found, however, that the data were not sufficient for his purpose, and it would be a matter of great interest to him if he were privileged to examine Mr. Freeman's experimental results.

In reply to Mr. Dodge, it must be a satisfaction to all concerned with the subject of pipe resistance that the results of his wide experience with steam agrees so well with those obtained by Saph and Schoder for water, and by Stanton and Pannell for both water and air. We may safely conclude that the basis of physical ideas from which the dimensional treatment starts is sensibly correct and that no important element in the problem has been overlooked. The results obtained by dimensional reasoning are so instructive and the problem of pipe resistance so important that the confirmation which Mr. Dodge has given is a valuable contribution to the subject.

The example, brought forward by Mr. Nusim, of the practical utilization of the motion of dynamical similarity, is very interesting. The author's object in presenting his paper was to call attention to the method which, he is convinced, will in time come to be one of the engineer's handy tools, like the two laws of thermodynamics. But since he is aware that his opinion of the value of the method may be received somewhat sceptically by professional engineers, tes-

timony in its favor from one engaged in practical designing work is doubly welcome.

In reply to Mr. Hersey, mechanical engineering is an art, not a science; and the ability and imagination of the individual engineer will always be its most important element. But common sense tells the engineer to get as much outside help as he can in solving his problems, and one source of such help is physics. As Mr. Hersey points out, the aid to be got from physics does not consist merely in new determinations of physical constants or in experimental investigations of physical problems which are of special interest to engineers. It consists also in the systematic use of the scientific method of physics in analyzing problems, planning experiments, and coördinating known facts so as to bring to bear on any new problem all the available knowledge, of whatever sort and wherever obtained, which may seem to be pertinent. This is the technical physics which is destined not only to work in its laboratories on problems presented by engineers, but to be recognized as an inseparable companion of sound and progressive mechanical engineering.

ON THE LAWS OF LUBRICATION OF JOURNAL BEARINGS

BY M. D. HERSEY, WASHINGTON, D. C.

Member of the Society

IN order to establish a rational basis for bearing design, it would be desirable to have empirical equations, or curves, showing accurately and completely how the friction loss and load-carrying power of bearings depend on all the physical conditions governing the action of lubrication, including, of course, the size, shape, and fit of the bearing, the speed, degree of lubrication, properties of the lubricant, and characteristics of the cooling system. The problem of mapping out the laws of lubrication in this general way, whether by piecing together existing data or by making new experiments, is such a complicated one that it is worth while to stop and consider whether any general principles are available which may serve to simplify it. A recognition of the above facts has led to the present paper.

Relation of laws of lubrication to design. After the requirements of strength and rigidity have been met, there may remain a question as to length and diameter which must be settled by reference to the laws of lubrication. Evidently too short a bearing is in danger of abrasion, while too long a bearing entails needless dissipation of power.

Let the coefficient of friction, f, be defined by the equation

$$f=\frac{F}{L} \quad \ldots\ldots [1]$$

in which F is the frictional resisting force and L the load on the bearing perpendicular to its axis. Let the bearing pressure, p, be defined by the equation

$$p=\frac{L}{l\,D} \quad \ldots\ldots [2]$$

in which l is the length of the bearing and D the diameter of the journal. Let p_o denote the carrying power or greatest permissible bearing pressure. Then the shortest permissible length of a bearing, l_o, may be calculated from the equation

$$l_o=\frac{L}{D}\cdot\frac{1}{p_o} \quad \ldots\ldots [3]$$

while the power dissipated in this bearing at a speed of n revolutions per unit time will be

$$P=\pi\, D\, n\, L\, f \quad \ldots\ldots [4]$$

Equations [3] and [4] are purely formal and their practical use demands a knowledge of some relation

$$f=\phi\ (p,\ n,\ D,\ l,\ \text{etc.}) \quad \ldots\ldots [5]$$

between the coefficient of friction and all the physical quantities governing the action of lubrication; together with some relation

$$p_o=\psi\ (n,\ D,\ l,\ \text{etc.}) \quad \ldots\ldots [6]$$

between carrying power and the factors which it may depend upon.

Equations [5] and [6] symbolize the most important of the laws of lubrication needed in design; and they may be termed the *law of friction* and the *law of carrying power* respectively.

Definitions of friction and carrying power. Already the coefficient of friction has been defined as the ratio of frictional resistance to the load on the journal perpendicular to its axis; while carrying power has been defined as the greatest permissible bearing pressure.

A satisfactory definition of permissible bearing pressure demands a definite understanding as to the type of failure we wish to avoid. If we wish to avoid failure by seizing, we must investigate thermal expansion; if we wish to avoid failure by overheating the lubricant, we must investigate the temperature rise; if to avoid creating tension in the lubricant, we must investigate the pressure distribution; if to avoid simple abrasion, we must investigate the minimum film thickness. The last type of failure is the only type we shall undertake to discuss in this paper.

If c denotes the *radial clearance* or mean difference in radii between journal and bearing, while x denotes the *film thickness* or thickness of the film of lubricant at the point of nearest approach, the fraction $\frac{x}{c}$ may be called the *relative film thickness*. In this paper, we shall make it a matter of definition that all bearings are equally *safe* which are running with the same relative film thickness. Accordingly, carrying power may be defined as that bearing pressure which reduces the relative film thickness to some prescribed value $\left(\frac{x}{c}\right)$.

The determination of the laws of lubrication symbolized by equations [5] and [6] therefore simmers down to the investigation of the effect of various conditions on F and x.

Restrictions necessary to exclude unfamiliar phenomena. In order to narrow the problem down to as simple a one as possible we may impose the following restrictions:

1 The bearing must be in a steady state.
2 The lubricant must be homogeneous.
3 The bearing must be running below the critical speed at which eddy motion would be set up in the lubricant.
4 The effect, on the motion of the lubricant, of any other forces than hydrostatic pressure and shearing stress must be negligible.
5 The metal surfaces must always be separated by a film of lubricant which is thick enough to have the same mechanical properties it would have in bulk.

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6 There must be no resultant couple acting on the bearing in the plane of its axis.

Qualitative discussion of action of lubrication. It is now shown qualitatively that, under the foregoing restrictions, F and x are completely determined by the following physical quantities:

a The viscosity, μ, of the lubricant.

b The revolutions per unit time, n.

c The load, L.

d The degree of lubrication; which, in the case of *stationary* lubrication (i.e., the limiting case when no lubricant enters or leaves the bearing) may be specified by V, the volume of lubricant in the bearing; and which in the case of forced lubrication may be specified by the quantity, Q, of the lubricant flowing through the bearing in unit time. Let S denote the relative supply $\frac{V}{Dlc}$

e The absolute size of the bearing, which may be given by the diameter of the journal, D.

f The line of action of the load, defined by some length ratio r' such as the ratio or its distance from the middle point of the bearing, to the diameter.

g The shape of the bearing; specified by the relative clearance $\frac{c}{D}$, the relative length $\frac{l}{D}$ and such other length ratios r'', r''', etc., as may be needed to fix the shape of the oiling arrangements, deviation from circular section due to wear, departure from cylindrical form due to strain, and all other geometrical irregularities. Let r denote all the ratios r', r'', r''', etc.

The conclusion that F and x depend only on μ n, L, V, Q, D, $\frac{c}{D}$, $\frac{l}{D}$, r, may be symbolized by the equations

$$\Phi\left(\frac{F}{L}, \mu, n, L, V, Q, D, \frac{c}{D}, \frac{l}{D}, r\right)=0 \quad \ldots\ldots [11]$$

and

$$\Psi\left(\frac{x}{D}, \mu, n, L, V, Q, D, \frac{c}{D}, \frac{l}{D}, r\right)=0 \quad \ldots\ldots [12]$$

Derivation of the general form of the laws of lubrication by dimensional reasoning. An application of Buckingham's II-theorem to equations [11] and [12] throws them respectively into the forms

$$f=\phi\left(\frac{\mu n}{p}, \frac{D^2 n}{Q}, S, \frac{c}{D}, \frac{l}{D}, r\right) \quad \ldots\ldots [20]$$

and

$$p_o=\mu n.\theta\left[\left(\frac{x}{c}\right)_o, \frac{D^2 n}{Q}, \frac{c}{D}, \frac{l}{D}, r\right] \quad \ldots\ldots [21]$$

f and p making their appearance in place of F and L by virtue of equations [1] and [2]. The functions ϕ and θ are of course unknown, and remain to be determined by experiment.

Equations [20] and [21] correspond to equations [5] and [6] respectively, and contain the two laws of lubrication in their most general form. Now, any two bearings in which $\frac{D^2 n}{Q}$ has the same value, and in which S also has the same value, may be called *similarly lubricated.* Hence [20] and [21] are equivalent respectively to the following statements:

a In geometrically similar bearings which are similarly loaded and lubricated, the *coefficient of friction depends only on the single variable* $\frac{\mu n}{p}$

b *The carrying power of any bearing is directly proportional to the product of viscosity by revolutions per unit time,* the constant of proportionality being the same for all geometrically similar bearings which are similarly loaded and lubricated and which are equally safe, i.e., $\left(\frac{x}{c}\right)_o$ constant.

The writer made a series of experiments on journal friction and carrying power several years ago, the results of which are entirely consistent with the above conclusions.[1]

Dynamically similar bearings. Any two geometrically similar bearings B and B' which are similarly loaded and lubricated, and which are running at the *corresponding* speeds, pressures, and viscosities defined by the equation

$$\frac{\mu n}{p}=\frac{\mu' n'}{p'} \quad \ldots\ldots [30]$$

have the same coefficient of friction and the same relative film thickness. Such bearings may be termed *dynamically similar.* The power dissipated in either of them may be calculated from a test made on the other, for

$$\frac{P}{P'}=\frac{D}{D'}.\frac{n}{n'}.\frac{L}{L'} \quad \ldots\ldots [31]$$

Moreover,

$$\frac{p_o}{p'_o}=\frac{\mu}{\mu'}.\frac{n}{n'} \quad \ldots\ldots [33]$$

Thus if the carrying power of one bearing has been found experimentally, that of the other can be at once calculated.

A general method for determining thermal effects. The principal effect of temperature, and the only effect we need analyze here, is to decrease the viscosity of the lubricant as the bearing heats up. Consequently both friction and carrying power increase with speed less rapidly, under working conditions, than they would at constant temperature. The general dynamical equations [20] and [21] are true regardless of temperature, because they have been expressed in terms of the actual viscosity of the lubricant in the film at the moment in question; but it is desirable to go a step further. Equations like [20] and [21], which describe the behavior of a bearing when the viscosity is given, may be termed *characteristic equations* for that bearing. On the other hand, an equation describing the behavior of a bearing, not in terms of the instantaneous viscosity, but entirely in terms of known constants or controllable conditions like speed and load, may be termed a *working equation* for that bearing. Such an equation does not characterize the bearing in itself, but depends also on the nature of the lubricant and on the cooling system. A general method for determining the working equations for the friction and carrying power of any bearing is now outlined. The method consists in eliminating viscosity from the characteristic equations by utilizing information about the lubricant and the cooling system. Five relations are available, namely those connecting the quantities

p_o and μ
μ and t
t and H
H and f
f and μ

respectively.

The first of these is the characteristic equation for carrying power, equation [21].

Any empirical equation for the viscosity of an oil in terms

[1] For a brief account of the experiments, see Jour. Wash. Acad. Sci., v. IV p. 549, 1914.

of temperature must involve certain constants, which are different for different oils; if we denote all these by a, the second relation can be written

$$\mu = F_1(t, a) \quad [34]$$

in which the function F_1 is supposed to have been determined empirically.

Likewise the heat carried away in unit time at any temperature t may be expressed by some empirical equation

$$H = F_2(t, b) \quad [35]$$

in which b denotes all of the constants entering the function F_2.

The fourth relation is obtained by equating J, the mechanical equivalent of the heat carried off in unit time, to the power dissipated according to equation [4].

The fifth relation is the characteristic equation [20] for the coefficient of friction.

Eliminating μ and H from the five relations leads to the three general equations

$$p_o = F_1(t, a) \cdot n \cdot \theta\left[\left(\frac{x}{c}\right)_o, \frac{D^2 n}{Q}, R\right] \quad [37]$$

$$F_2(t, b) = \frac{\pi}{J} D^2 l n p \cdot \phi\left[\frac{F_1(t, a) \cdot n}{p}, \frac{D^2 n}{Q}, R\right] \quad [38]$$

$$f = \phi\left[\frac{F_1(t, a) \cdot n}{p}, \frac{D^2 n}{Q}, R\right] \quad [39]$$

in which R has been written for all the ratios $S, \frac{c}{D}, \frac{l}{D}$, and r. These three relations may be regarded as a formal statement of the proposed method for determining thermal effects. The functions θ and ϕ are to be found by dynamical experiments; F_1 and F_2 by thermal experiments. After they have been determined, we can deduce a working equation for the carrying power by eliminating t from [37] and [38]. Likewise, by eliminating t from [38] and [39], we can deduce a working equation for the coefficient of friction. From [38] alone we get an equation for the permanent running temperature of the bearing.

If any of the empirical functions θ, ϕ, F_1, or F_2 prove too complex to represent analytically, we can still accomplish the desired eliminations graphically.

The relations [37], [38], and [39] are perfectly general and not limited to any particular type of bearing or to any particular lubricant or cooling system.

Properties of the Ideal Bearing. The paper closes with a discussion of the *ideal bearing*—one which is perfectly circular in cross section, completely filled with lubricant, etc. While intended primarily as an illustration of the foregoing relations, and not for actual use in design, the equations of the ideal bearing doubtless afford an approximation to the laws of actual bearings. The characteristic equations for the coefficient of friction and carrying power of a *high speed* bearing, or ideal bearing with approximately concentric journal, are respectively

$$f = \phi\left(\frac{\mu n}{p}, \frac{c}{D}\right) = \pi^2 \frac{D}{c} \frac{\mu n}{p} \quad [43]$$

and

$$p_o = \theta\left[\left(\frac{x}{c}\right)_o, \frac{c}{D}\right] \cdot \mu n = A \ \mu n \quad [44]$$

in which the constant A is supposed to have been determined empirically.

The following are convenient approximations for the viscosity and cooling functions respectively:

$$\mu = F_1(t, a) = \frac{\mu_o (t_o - \tau)}{t - \tau} \quad [47]$$

and

$$H = F_2(t, b) = h (t - t_o) \quad [48]$$

Here μ_o is the viscosity at the room temperature t_o; τ is the fictitious solidifying temperature; h the heat carried off in unit time per unit temperature elevation above room temperature.

If in [37], [38], [39], we now substitute the expressions for ϕ, θ, F_1, F_2, given respectively by [43], [44], [47] and [48], the following working equations result:

(a) For the permanent running temperature:

$$t = \tfrac{1}{2}(t_o + \tau) + \tfrac{1}{2}(t_o - \tau)\sqrt{1 + k n^2} \quad [59]$$

in which

$$k = \frac{4 \tau^2}{J} \cdot \frac{\tau_o}{h (t_o - \tau)} \cdot D^3 \cdot \frac{\left(\frac{l}{D}\right)}{\left(\frac{c}{D}\right)} \quad [60]$$

The quantity k may be termed the *heating constant;* the greater it is, the hotter the bearing will run at any given speed. Evidently k may be determined from an observation of the permanent running temperature at any one speed, for by solving [59] for k we see that

$$k = \frac{\left(\frac{2t - t_o - \tau}{t_o - \tau}\right)^2 - 1}{n^2} \quad [61]$$

(b) For the coefficient of friction:

$$f = \pi^2 \frac{D}{c} \frac{\mu_o n}{p}\left[\frac{2}{1 + \sqrt{1 + k n^2}}\right] \quad [62]$$

(c) For the carrying power:

$$p_o = A \ \mu_o n \left[\frac{2}{1 + \sqrt{1 + k n^2}}\right] \quad [63]$$

The above results apply only to the high speed bearing, for it is only when the journal is concentric that [43] is valid. The complete paper, however, contains a corresponding treatment of the ideal bearing with eccentric journal. Sommerfeld's work is made use of in this connection, and extended by an analysis of heating effects.

The paper contains illustrative examples, worked out numerically, and accompanied by diagrams.

Conclusion. It is apparent that the next step should be a minute examination of existing data, in the light of the foregoing principles. If it then appears that further experiments are needed, we may be guided as to the most economical way to plan them by reference to the general dynamical relations [20] and [21], and to the general thermal relations [37], [38], [39]. The dynamical relations serve to diminish the number of independent variables entering the characteristic equations. (Thus, the effect of varying all the factors μ, p, n, D, Q, can now be attained by changing two alone, such as μ and Q.) The thermal relations serve to diminish the number of combinations of conditions needed in determining working equations. It is a confusing task to map out the working equations of bearings under any great variety of circumstances by direct experiment, but a relatively simple matter to determine the dynamical characteristics of a bearing (ϕ and θ), and the thermal properties of a lubricant (F_1), and of a cooling system (F_2), by sepa-

rate sets of experiments. (Thus, in certain cases the function F_1 might be determined with a stationary dummy bearing, in which the heat is generated, regulated, and accurately measured electrically). It appears from this paper that it will be permissible to conduct such experiments separately; after which the results can readily enough be combined by the general thermal relations [37], [38], [39].

Finally, the conception of dynamically similar bearings may make it possible to evaluate some of the constants needed in design by the use of models—meaning of course by the model, any bearing whatever, found satisfactory in actual service, and which meets the condition of being dynamically similar to the bearing about to be designed.

DISCUSSION

H. F. MOORE. The paper appeals to the writer as an excellent example of the use of mathematical reasoning to facilitate and interpret experimental work. Coöperation between the mathematicians and experimenters is one of the most desirable technical achievements; many a carefully made set of experiments has gone for naught because of lack of analytical interpretation of results, or of proper theoretical lay-out of the tests, and many careful mathematical analyses have been valueless because the fundamental constants were wrongly assumed or not determined with sufficient accuracy.

The author's results on carrying power are qualitatively confirmed by the writer's experience. Experiments made by him in 1903 with a bearing of babbitt metal on a steel journal showed that the carrying power of the bearing before the film of oil broke down increased with some function of the speed, and so far as the tests went the bearing power was found to vary approximately with the square root of the speed. Later tests made at the University of Wisconsin on very carefully ground hardened steel journals rotating in bronze bearings showed a carrying power two or three times as great as did the babbitt metal bearing with unhardened journal. It might be expected that surface finish of journal and bearing would play a very important part in determining the breaking strength of the oil film.

W. H. HERSCHEL. The author's work is a notable step in bringing order out of the chaos of experiments on the friction of journals. If it can be demonstrated that viscosity is the only property of the lubricant which influences the coefficient of friction and the maximum permissible load, lubricating problems will be greatly simplified. Thanks to Sommerfeld, we no longer feel the necessity of considering "adhesion" which was introduced into equations by the earlier investigators, but there is a widespread belief that lubricants vary in regard to "oiliness," "lubricating value" or "body."

F. ZUR NEDDEN contributed a written discussion in which after comparing the author's results with recent experiments carried out by Professor Guembel in Germany and described in the Monatsblätter des Berliner Bezirks-Vereines Deutscher Ingenieure, May and June, 1914, he continued:

When gradually reducing (at constant pressure p) the speed of any bearing the value of the coefficient of friction in the neighborhood of zero, after reaching a minimum, abruptly increases to a high figure, because with very low speeds the liquid friction is giving way to semi-dry friction. The laws as set forth in the paper do not then hold good. The point where this change occurs depends among others on the smoothness of the gliding surfaces. This consideration is by no means only of theoretical interest. Toothed wheels when running in oil will show lesser friction losses if designed so that the flanks of the teeth glide on each other with a speed exceeding the upper limit of semi-dry friction.

The all-important question whether equations (11) and (12) are qualitatively complete should be further investigated in connection with a long series of very important tests which Professor Schlesinger has brought forth in contention of the idea that it is not correct to assume that the only factor that matters in a lubricant (besides its price) is its viscosity.

The writer has come to the conclusion that two properties of the lubricant probably have something to do with the process of lubrication, both of which are not covered by the author's formulæ. The first and less important one is the heat capacity of the oil itself, i.e., its specific heat. The second and very important one is the degree of amorphity. Lubrication depends entirely on the wedgelike action of the lubricant film. If the resistance of the lubricant against the mutual dislocation of its molecules is not absolutely the same regardless of the direction of the outer forces which tend to dislocate them, in other words, if the lubricant can no longer be regarded as a real amorphous liquid, then the application of Newton's law of viscosity will have to be modified.

Apart from optical reasons the change in the lubricating qualities of oils after some period of running would indicate that molecular forces are at work within lubricants which affect their action while not appreciably affecting their viscosity as measured by one of the ordinary viscometers.

Several very important conclusions can be drawn by the practical engineer from the results in this paper. One of the stipulations which is to be found in almost every specification for turbo-machinery is that the length of the bearings should at least equal three times the diameter. This rule of thumb probably is the outcome of practical experience gained with bearings of the comparatively low speed machinery exclusively used before the advent of the steam turbine. From the paper it follows the carrying power is different at different speeds.

The viscosity of the oil and the heat carrying capacity of the bearing are the essential standards which should be determined, but which are scarcely ever mentioned in any specification.

Of these two factors one depends on the buyer who supplies the lubricant for the machine he buys. Only the second factor, the heat carrying capacity, depends on the make. It is highly gratifying that the author shows a way to determine this quality of a bearing in a manner which will permit of standardization.

Until definite standards are developed, it would appear the safest way for a buyer to ask the maker for a guarantee of the temperature of the bearing, or rather of the maximum difference between room temperature and bearing temperature, provided such and such an oil is used.

In conclusion the writer considers the engineers of this country owe a very great debt of gratitude to the author for this valuable paper, and expresses the hope that his endeavors to map out the proper and most direct course for

the establishment of a rational basis for bearing design will be rewarded by a corresponding organization of the necessary experiments. If all individuals and laboratories, especially those of the Universities, who at present contemplate experimenting on the problem of lubrication of bearings could come together and assign among each other in a systematic and efficient manner the treatment of the various subproblems as indicated by the present paper—then progress in the knowledge of bearing construction should be quick and radical.

THE AUTHOR. In discussing the value of coöperating with mathematicians, Professor Moore evidently refers to mathematics in the customary sense in which the term is stretched to include physics as well, but the writer would urge that we break this custom, and always distinguish physical reasoning from mathematical reasoning. The distinction may seem hair-splitting at times, and yet we all know what havoc has been wrought in the field of elasticity, for example, by elaborating mathematically relations physically unwarranted. Aside from the Π-theorem, which owes its value solely to its capacity for summing up physical facts, there is no mathematics at all in the important part of my paper, namely, the first twenty-five pages. What looks to be mathematics is merely symbolism—shorthand. The conclusion that a bearing can support a heavier load the faster it runs, follows from the physical facts about viscosity; and the general method for grinding out working equations is merely a condensed statement of a group of facts about heating and cooling, familiar enough to physicists, but which the cleverest mathematician alive might not have heard of.

Professor Moore's own formula, the square root law of carrying power, has been unwarrantably extended by others, and applied to circumstances physically different from those under which it was established. Nevertheless, the historical significance of his experiments of 1903 can hardly be overestimated. They were the first experiments aiming toward a direct determination of carrying power.

While experimenting with bearings in 1909 at the Massachusetts Institute of Technology it occurred to the writer to see whether this square root law or any similar relation held under more nearly practical conditions; that is, with a whole bearing instead of a half bearing, and at higher speeds. It turned out that under these circumstances complete film rupture never took place, but that the load producing some specified electrical resistance other than zero did increase with speeed.

As noted by Professor Moore it is to be expected that (except at high speeds), bearings with differently finished surfaces will behave differently; they cannot be considered geometrically similar unless they are either sensibly smooth or similarly rough—not equally rough, but twice as rough if twice as big.

Mr. Herschel alludes to the widespread belief that oils of the same viscosity may still differ in lubricating value. This belief is doubtless correct as regards bearings running at low enough values of $\frac{\mu n}{p}$.

It would not be proper to comment in detail on Professor Guembel's work until Mr. zur Nedden's own discussion is printed in full. One fact may, however, be recorded for the interest of those who consult that reference in the meantime: If the curves on which Professor Guembel's final equations are based be fitted at three points instead of two, a more general type of equation results, which reduces to Guembel's at low values of $\frac{\mu n}{p}$ and to Sommerfeld's at high.

It is true that the present paper is inadequate to cope with semi-dry friction, and it is likely that Professor Schlesinger's experiments, dealing as they do with the aggregate friction losses of the engine lathe, are largely concerned with semi-dry friction.

The specific heat of the oil has to be regarded as one of the b's of equation [35] for heat carried off.

The writer does not believe there is, as yet, any need for considering the effect of incomplete amorphity.

Mr. zur Nedden's suggestions are exceedingly interesting, and must certainly be taken account of in the subsequent development of the subject. But, for the moment, we are so much at sea that it is well to follow the rule of navigation and keep to deep water, in doing which our six fundamental restrictions will serve as the channel buoys. Any exact charting out of the shoals has, to be sure, been postponed.

INFLUENCE OF DISK FRICTION ON TURBINE PUMP DESIGN

BY F. ZUR NEDDEN, NEW YORK, N. Y.

Member of the Society

INTRODUCTION

A DISK revolving in water acts as a brake. The losses due to the rotation of impellers in the water surrounding them amount t oone-quarter to one-third of the total of the losses occurring in a high-lift turbine pump. It is astonishing that publications on the nature and magnitude of these losses are more scarce.[1, 2, 3, 4] There is very little reliable information on the question and that little is not ararnged as designers would like to hav it for us in commercial practice.

In the present paper, the results gained by Prof. A. H. Gibson[4] are mainly employed in an effort:

a To find what means are at the disposal of the designer of turbine pumps for minimizing the losses due to disk friction.

b To furnish reliable data and diagrams from which to estimate the influence of disk friction on the efficiency of a turbine pump.

c To draw attention to the bearing which disk friction of the impellers has on the axial thrust of turbine pumps.

NOTATION

κ_A and κ_B = coefficients of friction between surfaces A and B respectively on one hand and the fluid surrounding the disk on the other hand.

Waste-water = fluid surrounding the disk.

f = Gibson's coefficient of disk friction.

v_r = abs. velocity of waste-water in ft-sec. at the distance r from the axis.

[1] "Experiments on the Friction of Disks Rotated in Fluid." Minutes Proc. Inst. Civ. Eng., London, vol. lxxx, p. 221 ff.
[2] Bulletin No. 2, Univ. of Cal., Berkeley, Cal., 1887.
[3] Wirkungsweise der Kreiselpumpen; Mitteilungen ueber Forschungsarbeiten, Verein Deutscher Ingenieure, Heft 42, Berlin, 1907.
[4] Min. Proc. Inst. Civil Eng., London, vol. clxxix, 1910, part 1.

Presented at the Spring Meeting of the Society, June, 1915.
Complete paper may be obtained without discussion; 20 cents to members; 40 cents to non-members.

v_ρ = absolute velocity in ft-sec. of a point situated on the surface of the disk at a distance of ρ from the axis.

$v_{\rho A}$ = abs. velocity in ft-sec. of adjacent particle of waste-water.

$v_{\rho B}$ = abs. velocity in ft-sec. of waste-water particles adjacent to casing at distance ρ from axis.

O_A = area in sq. ft. of that part of the impeller surface which is marked by fat contour in upper half of Fig. 1.

O_B = area in sq. ft. of surface of casing as marked by fat contour in upper half of Fig. 1.

P = dragging force in pounds of impeller, i.e., force exerted by impeller on waste-water which it causes to rotate.

Q = retarding force, in pounds, of casing, resisting effects of P.

ω_T = angular velocity per sec. of waste-water.

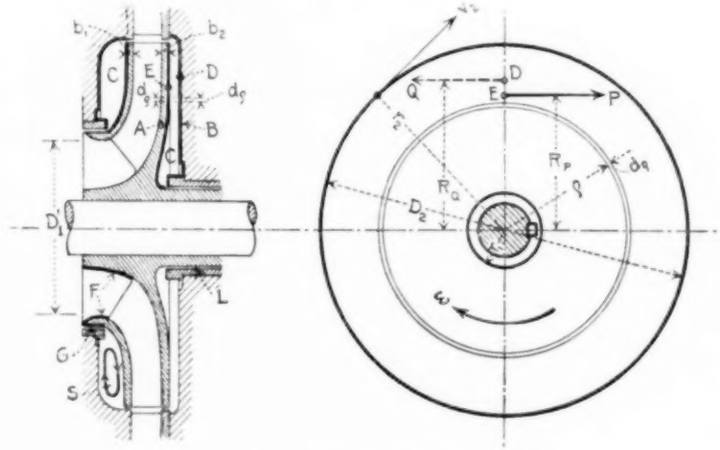

FIG. 1 KEY TO NOTATION

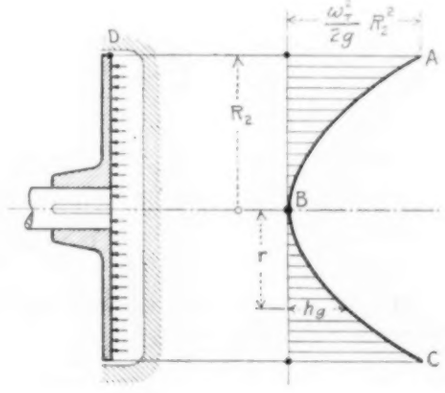

FIG. 2 GYROSTATIC PARABOLA

ω = angular velocity per sec. of impeller.

J_A = polar moment of inertia in ft⁴, of surface O_A with respect to axis of rotation.

J_B = the same of surface O_B

R_p and R_q = distance, in feet, of P's or Q's place of application respectively from axis of rotation.

E_r = energy of disk friction in ft-lb.

h_g = gyrostatic head in ft.

h_s = static head generated by impeller proper in ft.

D_1 = diameter of slip ring clearance in ft.

D_2 and R_2 = diameter and radius respectively of impeller in ft.

P_g = total gyrostatic force in lb., see equation [9].

P_s = total axial force exerted by gyrostatic plus static pressure on the impeller in lb., see equation [10].

N_T = number of revolutions of waste-water per min.

N = number of revolutions of pump per min.

After pointing out the conditions when and where the disk friction may be expected to be the smallest possible, the author makes the following deductions:

a In order to reduce the loss through disk firction as much as possible, the coefficients of friction κ_A and κ_B should be made as small as possible; i.e., both the casing and the runner should be machined or polished as smoothly as possible.

b If a smoothly finished impeller revolves within a rough casing of a lateral width not exceeding about 5 per cent of the diameter of the impeller, the result is the same as if a rough impeller revolves within a smooth casing.

c The surface of the impeller and of the stationary parts of chamber C (Fig. 1) should be made as small as possible.

d The effect of an extended and complicated surface is equally bad whether this surface be the stationary wall of chamber C or the rotating surface of the impeller.

The outward indication of the attainment of a minimum of loss through disk friction is the fact that the waste-water rotates just half as quickly as the impeller.

THE GYROSTATIC PRESSURE

The pressure due to the rotation of the waste-water is the source of important axial forces. At a distance of r ft. from the axis of rotation and an angular velocity of ω_T per sec. this pressure in feet of fluid is

$$h_g = \frac{\omega_T^2 r^2}{2g} = \frac{v_T^2}{2g}$$

where v_T stands for the absolute tangential velocity of the waste-water in ft-sec. and $g = 32.16$ ft-sec.

When plotting these pressures as abscissæ over the respective radii as ordinates, the parabola $A\ B\ C$, Fig. 2, is obtained. The pressure h_g is directed towards the circumference.

In turbine pumps the problem is to find the influence of the gyrostatic pressure upon annular surfaces, say, e.g., that part of the impeller which lies between slip ring a and its periphery (Fig. 3).

The gyrostatic force in pounds axially exerted by the rotating waste-water upon an annular surface is shown in this section to be

$$P_g = \frac{N_T^2 (D_2^4 - D_1^4)}{1000} \quad \ldots\ldots\ldots\ldots [9]$$

where

N_T = rotative speed of waste-water in r.p.m.

D_2 and D_1 = the outer and the inner diameter, respectively, of the surface in ft.

and the total axial force in a pump which generates static pressure.

$$P_s = 49(D_2^2 - D_1^2)\left(h_s - \frac{4.25 N_T^2 D_2^2}{100{,}000}\right) + P_g \quad \ldots\ldots [10]$$

where

D_2 and D_1 = outer and inner diameter, respectively, of annular face, in ft.

h_s = static head existing at circumference of impeller, in ft.

N_T = rotative speed of waste-water in r.p.m.

P_g = gyrostatic force (see equation [9]).

The paper here surveys the experiments of Unwin and Gibson, considering the limitations of their formulae and also that of Biel for the theoretical loss through disk friction. It then develops the following expression:

$$h.p.f' = \frac{4\pi^{n+2} f}{550.30^{n+1}.2^{n+3}} N^{n+3} \frac{D_2^{n+3}}{n+3} \quad \ldots\ldots [12]$$

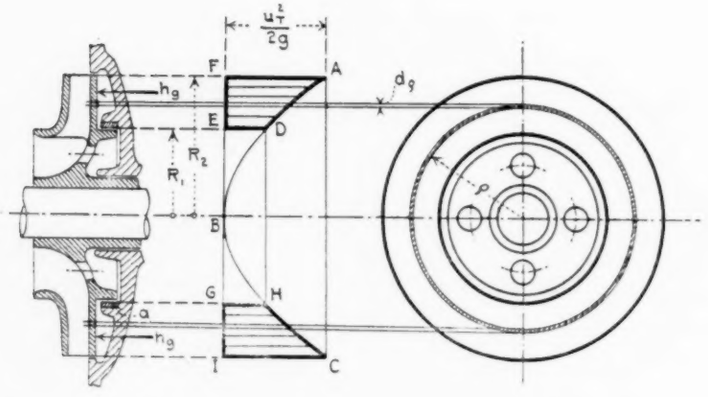

FIG. 3 GYROSTATIC FORCE ACTING UPON ANNULAR SURFACE

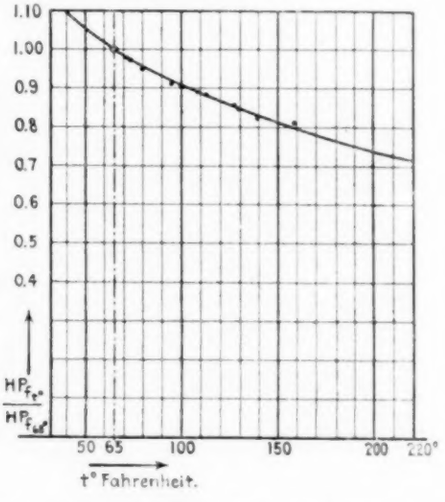

FIG. 4 DISK-FRICTION LOSS AS FUNCTION OF TEMPERATURE OF WASTE-WATER

where

$h.p.f$ = loss due to friction on both faces of disk, without considering axial thickness, in h.p.

f = coefficient of friction, see Fig. 5, *a*, *b* and *c*.

n = exponent of friction, see Fig. 5, *a*, *b* and *c*.

N = r.p.m. of disk.

D^2 = largest diameter of disk.

This formula does not take into account the axial extension of the disk and so the author develops a corrective factor for use of designers, viz.:

$$h.p.f_b = h.p.f' \times \left[\frac{b}{D_2}.(n+3)\right] \quad \ldots\ldots [14]$$

That is, in order to find the total loss due to *friction both of the circumferential and the lateral faces* of a disk, the value of $h.p.f'$ obtained for the latter (equation [12]) should be multiplied by

$$1 + \frac{b}{D_2}(n+3) \quad \ldots\ldots [15]$$

where b is the total axial thickness of the disk, expressed in the same measure as D_2.

In this way the designer can always, by a simple factor to be worked out mentally, consider separately and keep apart the influence of the diameter and that of the axial extent of an impeller on the friction loss.

INFLUENCE OF ROUGHNESS OF DISK AND CASING

Figs. 5 *a*, *b* and *c* represent graphically the average variations of f and n in function of r_2 as established by Gibson and Ryan for various degrees of roughness. Their results were compiled with those established by Professor Unwin's experiments and found to harmonize with them to a satisfactory degree. As set forth in Figs. 5 *a*, *b* and *c* the values of f and n furnish, within the range of speeds at which turbine pumps are run, figures for the losses (equations [12] and [14]) which always come within 5 per cent of the actual test results.

The second deduction at which the author arrived by a mathematical survey in this paper, viz.: "If a smoothly finished impeller revolves within a rough casing, the result is the same as if a rough impeller revolves within a smooth casing," is well corroborated by these curves.

The effect of metal polish and of varnish is amost exactly the same. This result should, however, not encourage manufacturers to restort to varnishing rather than machining the surfaces. Varnishing or japanning appears only at first sight to be the cheaper process. As a matter of fact, two coats of varnish require rather a long time to dry thoroughly. If the varnish or lacquer is not applied very carefully and dried well, or if the fluid to be pumped is not absoiutely clean, the varnish will peel off. The surface then acts worse than even a rough casting. Polished surfaces, on the other hand, will often lose much of their original finish by the sediments of the water in the pump. It is, therefore, of no great use to expend much time and wages on a high, shining exterior finish of the impellers. It is sufficient to turn the faces off with a medium heavy cut, especially as the turning ruts are running concentric, i.e., in the direction of the flow of the waste-water.

Much saving can be effected if designers and shops coöperate in producing the smoothest possible castings to surround the waste-water chamber. This chamber should offer a minimum of surface and be free of any ribs, protrusions or re-

cesses which not only increase the surface and impede the rotation of the waste-water, but also frequently give rise to rough castings. If the designer gives the walls of the chamber a shape of utmost simplicity, the foundry will be able to produce faultless, smooth castings without extra cost.

Considerable though the differences due to various degrees of ordinary roughness in new pumps may at first seem, still it is only with typical high lift pumps that they appreciably affect the efficiency of the pump. This is especially true where the duty is small in comparison with the head per stage.

The paper here gives an example to illustrate this thesis.

INFLUENCE OF VISCOSITY, TEMPERATURE AND SPECIFIC GRAVITY OF FLUID ON DISK FRICTION

Inasmuch as oil, gasoline, hot water, etc., are today pumped by turbine pumps, the consideration of these influences gains (65 deg. fahr.) is eventually used, say, as a boiler feed pump and deals with water 200 deg. fahr.

Of course, in a pump all the other causes inherent to the change of temperature are at work at the same time. For instance the loss of head owing to hydraulic friction in the fixed and rotative channels of the pump is likewise diminishing. On the other hand, the leakage loss must increase as the fluid becomes less viscous. The balance of all influences tends slightly to improve the efficiency of a boiler feed pump in actual service as compared with the figures obtained when testing it with cold water. The paper gives two examples illustrating the influence of heavy oil and tarry liquid on the disk friction in a turbine pump.

INFLUENCE OF DIAMETER AND SPEED

Formula [12] with $n = 1.8$ to 2, reveals at once the tremendous role the diameter plays. The influence of the rotary

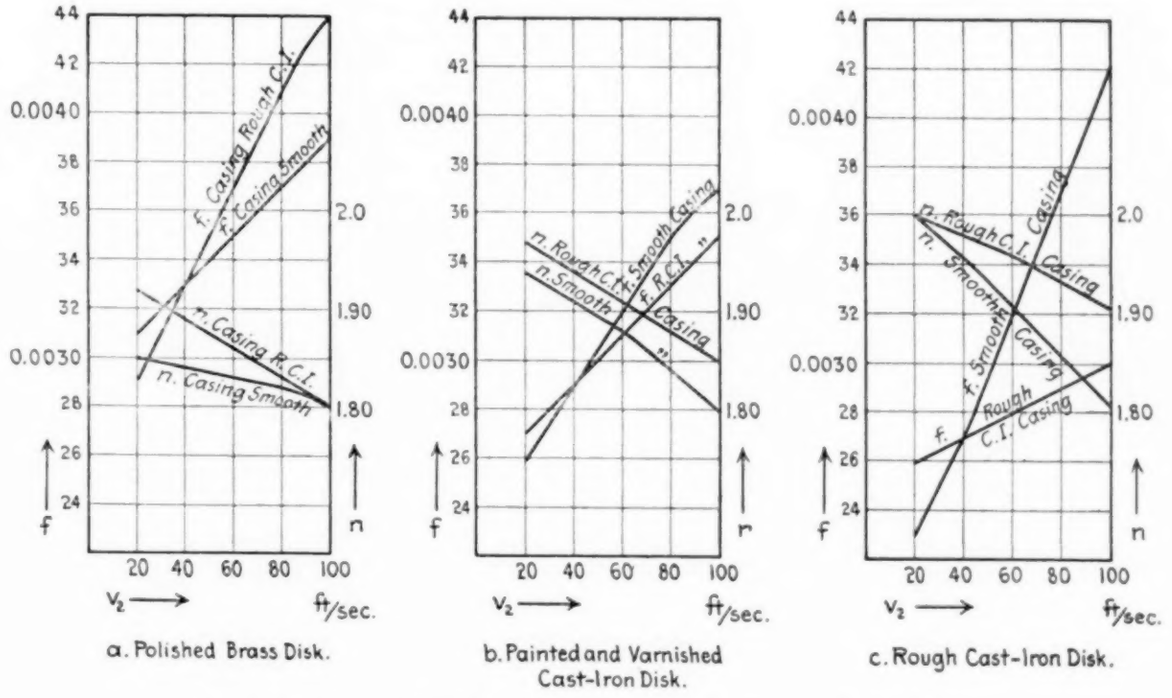

FIG. 5 DIAGRAMS OF DISK FRICTION ON POLISHED, PAINTED AND ROUGH SURFACES

in importance. The author has found it possible to answer the question of their respective magnitudes by a simple application of the theory of dimensions.

By considering the dimensions upon which the magnitude of the resistance depends, the expression

$$h.p.f' = k' \sqrt[5]{\frac{\mu . w^5}{s}}\ v^{2.5} d^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [23]$$

where μ = absolute coefficient of viscosity.
w = specific gravity.
s = lateral width of casing.

is deduced. It is easily seen that this is substantially identical with formula [12].

Fig. 4 represents the variation of *h.p.f* with the temperature of the water. This curve may be utilized for estimating the improvement in power consumption which may be expected when a pump which has been tested with cold water

speed N is of relatively minor importance, though it is in proportion to about the third power of the number of revolutions.

As the duty of a turbine pump is in proportion to the speed, and the head in proportion to its square, the useful output is increased in proportion to the third power of the speed. It will be seen, therefore, that the *loss through disk friction must form a constant percentage of the normal useful output, (water-h.p.), of a given pump irrespective of speed.*

High heads per stage are more economically produced by applying high speed than by using a large diameter of the impeller. This economic tendency for the high-speed pump is fostered by a number of other incidental advantages, e.g., alternating current electric drive, direct steam turbine drive, small unit costs, small weight, reduction in tensile stress on casing, compactness, ease in transport and repairs, etc.

It might not be out of place to point out that *the angle between the impeller blades and the tangent at the periphery* has an important bearing on the loss through disk friction. If this angle is made 20 deg. the diameter must, from hydraulic reasons, be larger by 10 to 25 per cent, than if the angle were 45 deg. This implies for the flatter angle an increase in disk friction of at least 60 to more than 150 per cent in some cases. As the disk friction loss expressed in percentage of the useful work amounts seldom to less than 5

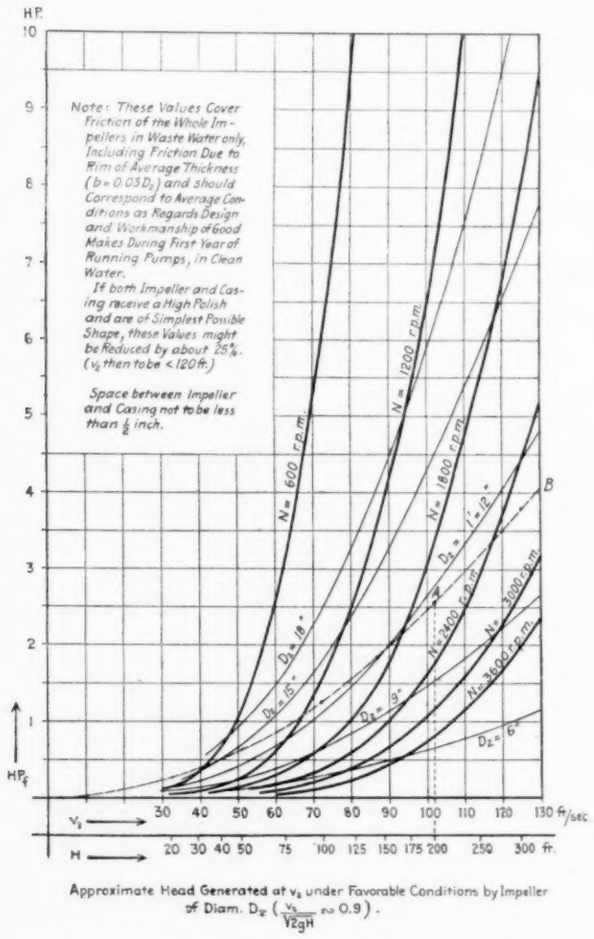
HP
Note: These Values Cover Friction of the Whole Impellers in Waste Water only, Including Friction Due to Rim of Average Thickness ($b = 0.03 D_2$) and should Correspond to Average Conditions as Regards Design and Workmanship of Good Makes During First Year of Running Pumps, in Clean Water.
If both Impeller and Casing receive a High Polish and are of Simplest Possible Shape, these Values might be Reduced by about 25%. (v_2 then to be < 120 ft.)
Space between Impeller and Casing not to be less than ½ inch.
N = 600 r.p.m.
N = 1200 r.p.m.
N = 1800 r.p.m.
N = 2400 r.p.m.
N = 3000 r.p.m.
N = 3600 r.p.m.
D_2 = 18"
D_2 = 15"
D_2 = 12"
D_2 = 9"
D_2 = 6"
B
HP_f
v_2
H
30 40 50 60 70 80 90 100 110 120 130 ft/sec
20 30 40 50 75 100 125 150 175 200 250 300 ft.
Approximate Head Generated at v_2 under Favorable Conditions by Impeller of Diam. D_2 ($\frac{v_2}{\sqrt{2gH}} \sim 0.9$).

FIG. 6 LOSS THROUGH DISK FRICTION IN FUNCTION OF DIAMETER AND SPEED, BASED ON FORMULAE 12 AND 14

and often to more than 10 per cent in high lift pumps, it is evident that the mere changing of the vane angle from 20 to 45 deg. implies a profit of from 3 to 8 per cent and even more in the efficiency of such pumps. The reason why many designers like to choose flat angles of about 20 deg. is that the water then leaves the runner at a relatively high static pressure and low speed, i.e., that little or no attention need be paid to the balance of the kinetic energy. Diffusors or guide wheels may be dispensed with, and eddies due to an improperly simplified design of the casing cannot do much harm. Still, competition in regard to efficiency between various makes and the urgent demand of the salesmen to reduce the motor power in order to minimize the cost of the unit to the consumer gradually force the manufacturer and designer to abandon their preference for flat angles and to utilize the advantage that results from the consequent reduction in diameter and disk friction.

In order to enable designers and manufacturers to estimate the loss due to disk friction without resorting to formula [12] which is rather unwieldy for practical use, the author plotted the diagram Fig. 6, which should be self-explanatory. In basing the chart on the highest values found by Gibson and adding 15 per cent for friction of rim (supposed to be 3 per cent of D_2 thick) the author felt he should come nearest to conditions as they obtain in practice.

The author here gives some examples serving to illustrate the value and the mode of using this figure: the following example is a typical one:

Suppose 250 gal. per min. were to be pressed against a total head of 600 ft. and the speed of about 1800 r.p.m. had to be adhered to, say on account of 60 cycle a.c. current motor drive. What number of stages should be chosen?

As 200 ft. per stage is too much for that capacity, either a 4-stage, a 5-stage or a 6-stage pump must be employed. By using Fig. 6 the following table (Table 1) can be compiled without difficulty:

TABLE 1

INFLUENCE OF NUMBER OF STAGES ON PERCENTAGE VALUES OF LOSS THROUGH DISK FRICTION

Q = 250 gal. per min. N = 1800 r.p.m.	Head per stage in ft.	Water h. p. per stage	Loss through disk friction	
			In h. p. per stage	In percentage of water h.p.
4-stage pump...	150	9.5	1.62	17
5-stage pump...	120	7.6	0.98	12.9
6-stage pump...	100	6.3	0.66	10.5

If the 4-stage and the 6-stage pump were designed equally well, the 6-stage pump would have an inherent advantage, due solely to the number of stages, of about 6.5 per cent of the water-horse power, or about 4 per cent in efficiency. The first cost would be increased in proportion to the number of stages, and it would be a question of the cost of energy to the consumer whether the 4 per cent advantage in efficiency would outweigh the excess in price of the 6-stage pump over that of the 4-stage pump.

Looking at the question from the manufacturer's point of view, it might be possible, by abandoning certain niceties in design, for instance, the expensive guide vanes, to turn out a 6-stage pump actually as cheaply as the more elaborate 4-stage pump. If more than 4 per cent of efficiency was not sacrificed by that simplification the 6-stage pump would be as advantageous as the 4-stage pump—leaving out the question of number of spare parts, dimensions and weight.

One more point comes in for consideration: an impeller designed for 250 gal. only and required to pump against a 150-ft. head per stage would have to be large in diameter. As the duty is comparatively small, the passages at its circumference become rather narrow—too narrow perhaps for the cores to be properly fixed in the mould, or the passages to be well filed out in the shops. For this reason alone the head per stage is limited. The admissible minimum number in the present instance is probably 5-stage. The competition then stands between the 5-stage and the 6-stage pump. It is evident that it will be a difficult decision whether the

slight increase in efficiency due to the one stage more will warrant the extra cost of that stage. Leaving out the possible advantages in manufacturing costs which might result from a greater output, a skilful designer may be quite able to surpass the efficiency of a competitive 6-stage pump with a 5-stage pump of like or even lesser cost per stage, though the former has a natural advantage of 2.5 per cent less in disk friction.

At any rate the question will be of such nicety as to make it impossible for the buyer, judging merely from the number of stages, to have a sufficient insight into the merits of a pump. *Specifications, therefore, should not fix the number of stages, but leave this point to the discretion of the manufacturer.*

INFLUENCE OF LATERAL DISTANCE BETWEEN CASING AND DISK

Fig. 7 represents a diagrammatic compilation of the results which Gibson and Ryan obtained when varying the lateral distance, s, with those found by Wagner when rotating a disk in practically an unlimited basin ($s = \infty$) and by Dr. Becker when rotating a polished piston within a very narrow polished casing.

From the diagram it is clear that the loss due to disk friction has, in general, a rising tendency with increasing width, s, of the waste-water chamber.

Nothing can illustrate better than this diagram the necessity of using every caution in deriving conclusions from any mathematical survey of hydrodynamic problems.

The designer will draw two conclusions from Fig. 7. First, there is in general a tendency for the disk losses to rise when the width of the waste-water chamber is increased.

Second, concentric circular protrusions or ribs are not objectionable and might even be of advantage by breaking the secondary currents; while radial ribs must increase the disk losses because they favor the generation of induced currents and additional losses caused thereby.

INFLUENCE OF LATERAL DISTANCE AND ROUGHNESS UPON THE AXIAL THRUST

Professor Gibson, when investigating the effects of radial vanes attached to the impeller, measured the pressures generated by them. Incidentally he made a few readings from the gage attached for that purpose at the casing (which had an internal diameter of 13 in.) while rotating smooth disks. The author found that these accidental by-products of Gibson's tests are of the utmost importance for explaining and, therefore, mastering some components of the axial thrust which for a long time remained inexplicable to him and to many others.

In Fig. 8 the lower three curves (curves of h_s) are plotted from Gibson's results. They represent the pressure difference due to the rotation of the waste-water between the center and points at a radial distance of 6½ in. from it.

From these readings it is possible to work out the rotary speeds of the waste-fluid, N_T, which, for the particular value of $v = 0.541$ ft. obtaining in Gibson's apparatus, is

$$N_T = 215\sqrt{h_s}$$

By far the most important consequences follow from Fig. 8 when used to estimate what axial pressures will result if the roughness of disk or casing, or their distance from each other, differ on the two sides of the impeller.

Equation [10] furnishes the total axial force resulting from the coöperation of the static pressure h_s generated by the impeller vanes at the circumference, and the gyrostatic pressure h_g. If the rotary speed of the waste-water N_T is different on the two sides of the same impeller an axial thrust results. This thrust is most striking with so-called balanced impellers, i.e., either double inlet runners, or runners having a slip ring on the back and borings leading from the inlet chamber through the eye into the chamber thus formed on the back (see Fig. 9). It shall be supposed that the static pressure generated by the impeller vanes is identical on both sides of the impeller. (This does not exactly hold true for single-inlet impellers for reasons which have no bearing upon the problem treated in this paper.)

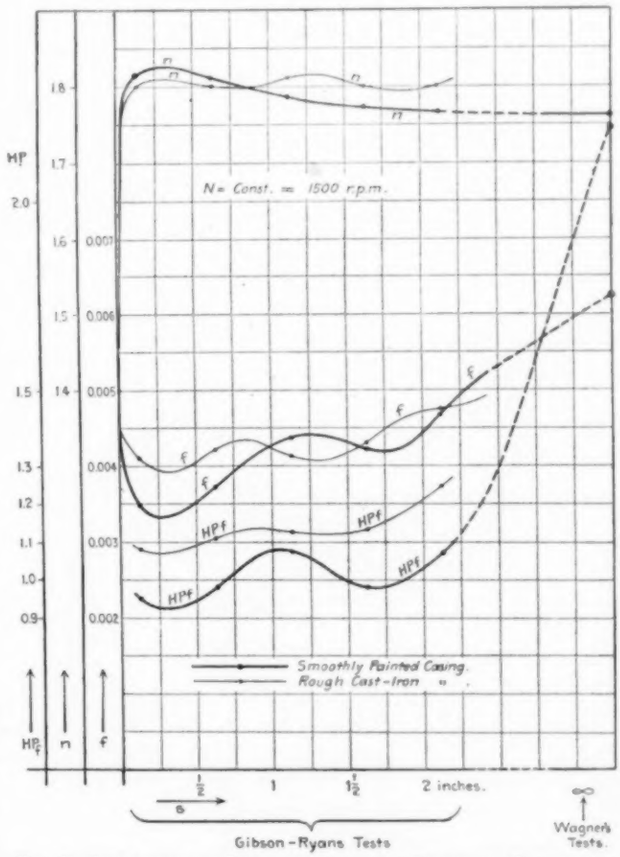

FIG. 7 INFLUENCE OF LATERAL DISTANCE BETWEEN CASING AND POLISHED 12-IN. DIAMETER DISK

When making the above assumption h_s (see equation [10]) is eliminated when subtracting the total axial force on one side from that on the other, and the effect of the gyrostatic phenomenon can be treated independently of any other causes which might affect axial thrust.

After subtracting the two values which equation [10] assumes for two different rotary speeds of the waste-water bodies on both sides of the impeller, the following equation results:

$$P_{s-s'} = P_g - P_g' - \frac{4.25 \times 49}{100000}(D_2^2 - D_1^2)D_2^2(N_T^2 - N_T'^2) \quad \ldots\ldots [25]$$

Introducing the values for P_g and P_g' which result from equation [9] this final expression is obtained for the differential axial force:

$$P_a = PL_{a-a'} = \frac{N_T^2 - N_T'^2}{1000} [2.08 (D_2^2 - D_1^2) D_2^2 - (D_2^4 - D_1^4)] \quad [25_a]$$

where

N_T and N_T' = r.p.m. of waste-water bodies on two sides of impeller.

D_2 = outer diameter of impeller in ft.

D_1 = diameter of clearance of slip-rings in ft.

P_a = axial thrust resulting from difference between N and N_T', in lb.

P_a' depends on the fourth power of the diameter. No appreciable inaccuracy is caused by disregarding the central part of the impeller.

Fig. 9 shows the direction of the resulting thrust P_a.

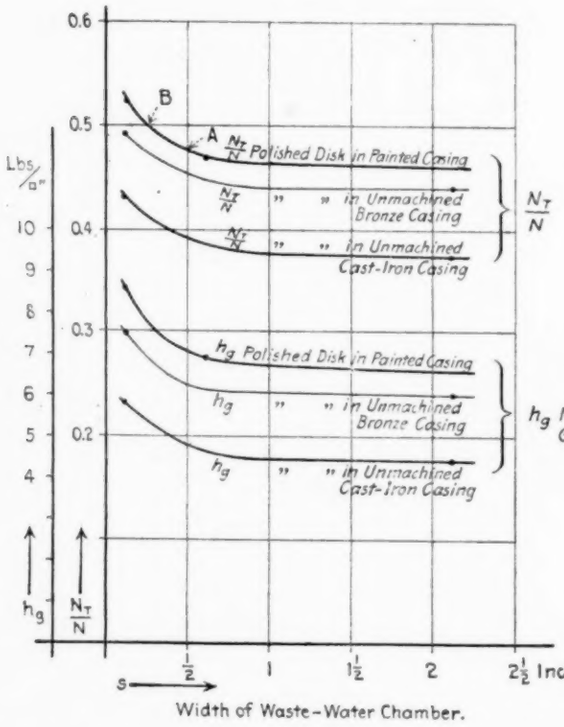

FIG. 8 ROTARY SPEED AND PRESSURE IN 13-IN. DIAMETER CASING PRODUCED BY 12-IN. DIAMETER DISK RUNNING AT 1200 R.P.M., IN FUNCTION WIDTH OF WASTE-WATER CHAMBER

The difference in pressure between axis and periphery is greatest where the waste-water rotates quickest (Fig. 9; right hand side—owing to small distance s). The static pressure h_s is the same on both sides. On the side where the waste-water rotates faster the gyrostatic pressure cuts out a greater parabolic area than at the other side. Therefore the total weight of pressure is smaller on the side where the waste-water rotates faster. Perhaps the following rule is easier to keep in mind: *"The impeller is drawn to the side where the waste-water rotates fastest."*

Two more examples, intentionally chosen to emphasize the necessity of the designer's attention to gyrostatic axial thrust, are here worked out in the paper. They show that an axial thrust of 440 lb. can be easily accounted for by the gyrostatic pressure differences forming in a 4-stage pump with impellers 1 ft. in diameter and the lateral width of the waste-water chambers ½-in. on one and ¼-in. on the other side of each impeller. Mention is made that further investigation of the details of the curves in Fig. 8 would be very desirable.

The gyrostatic axial forces, though important, do not constitute the only hidden cause of axial thrust. There are other variable components of the axial forces due, for instance, to the flow of the leakage water, or to the difference in pressure within the cells of the impeller, etc. Designers may never hope to obtain perfect balance under more than one particular condition or independent of wear and tear. They must always rely upon some strong hydraulic balancing mechanism or thrust bearing. The value of the above formulæ, therefore, lies principally in the possibility to foresee, by their aid, any danger that the axial thrust of the pump may act in the direction opposite or in addition to that for which the balancing device or thrust bearing is designed. It is known what strong axial forces can be exerted

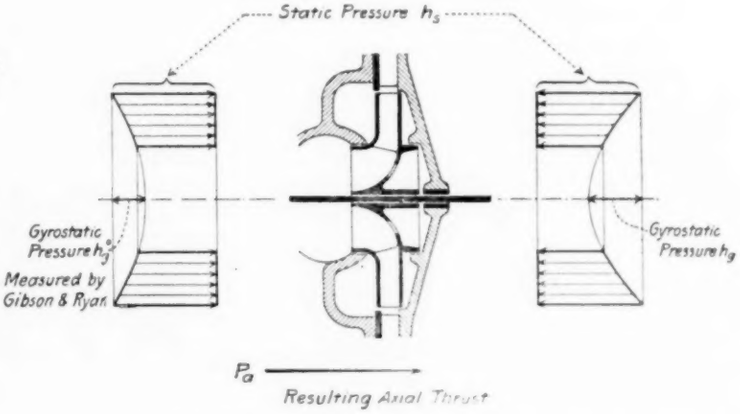

FIG. 9 DIAGRAM INDICATING AXIAL THRUST DUE TO EFFECT OF ROTATION OF THE WASTE-WATER

by disks provided with radial ribs. In this connection the results which Professor Hesse derived from investigating hydraulic footsteps offer some valuable data to the designer, as does the paper of Gibson and Ryan. Not so well known, but no less remarkable is the effect of radial vanes in the fixed walls of the waste-water chamber. From the mathematical survey in Section II it is evident that their effect must about equal that of rotary vanes. The author knows of the experiences of a well-known pump manufacturer who had provided radial ribs in one, but not in the other waste-water chamber of a perfectly symmetrical double inlet impeller. The strong axial thrust which resulted was reduced to almost nil after the pockets between the ribs were covered by means of sheet iron, and incidentally the efficiency was considerably increased.

In case a designer should find it difficult for general constructive reasons, to shape both waste-water chambers exactly alike, he will, on the basis of the information contained in this paper, be able to counterbalance the effect of any differences in roughness and shape. It is scarcely necessary for the author to elaborate on this question which must be solved in a different way in each individual case.

DISCUSSION

C. GEORGE DE LAVAL. The author draws the conclusion that effect of roughness of surfaces on impellers and internal parts does not to an appreciable degree affect the efficiency of the pumps. In late reesarches by the National Physical Laboratory in England, it has been shown that the change from lamellar motion of liquid to eddying motion takes place suddenly at a definite value of the critical velocity and which is directly proportional to the kinematical viscosity of fluid. The conclusions by Lord Rayleigh indicate that, by the principle of dynamical similarity, the relation of bodies immersed in fluids moving relatively to them is a general law of resistance of bodies and depends on the velocity, density and kinematical viscosity of the fluid expressed by the formula

$$R = \rho v^2 F\left(\frac{\mu d}{s}\right)$$

Where d = section or diameter of casing or pipe
R is resistance per unit area
S = specific gravity
F is a function of the one variable $\frac{\mu d}{s}$
ρ = density of water
v = mean velocity of water
μ = viscosity

The information given by the author is exceedingly interesting, particularly in the light thrown on fluids affected by viscosity, temperature and specific gravity. There is no doubt that little is known of this, not only on movement through pipes but also in chambers of centrifugal pumps. It is, no doubt, difficult to obtain a practical formula for use in dealing with very viscous liquids, nor can one estimate even approximately what can be expected in the way of power, speed and capacity.

The writer has found that results obtained are at considerable variance with what has been expected from mathematical conclusions based on existing laws of water. It would be most helpful if a commission were appointed to conduct a series of experiments to determine frictional resistances of oil and heavy viscous liquids in pipes, centrifugal pumps and reciprocating pumps and so finally establish a practical formula which could be relied upon. The oil industry is so great and important that more research information should be made available to the engineering profession than is now. Today we have not even a standard viscometer which is satisfactory; nor is a final method settled upon as to basis of figuring results and comparing with water.

The guaranteed performance of a centrifugal pump for oil is always based on clear water, the friction of oil in pipes and pump being assumed on this basis. The final results in pumping oil with a specific gravity of 0.87 show an increase of power of about 20 per cent and a reduction in head of about 20 per cent. The internal disk friction of impellers on account of viscosity of the oil causes the efficiency to drop 25 per cent. Theoretically the pumping head, whether oil or water, should be the same, but owing to the viscosity of the liquid there is considerable heat generated by the impellers in the pump chamber and the friction loss of the liquid increases, due to the viscosity in impeller passages, so that the head produced is about 20 per cent less than with water. It is, therefore, necessary that experiments are resorted to as a considerable number of errors will creep in and give misleading results.

The author claims that he answers these important questions by theory without resorting to experiments. To the writer this appears a rather dangerous proceeding as to influence of viscosity, temperature and specific gravity of fluids on disk friction. Theories should be proved out by experiments. The work wasted in impeller friction, which is a wasted power due to skin friction, is the most serious loss and can be obviated by having the impeller revolve in atmospheric air, sealing the edges of outer circumference of impellers at each side of the rim and also at hub; this would go a long way towards increasing over-all efficiency and reduce the power. Any leakages at these points can be automatically taken care of, allowing this water to go back to its original source, the amount of which would be very small.

Impellers could be made also to operate in air pressure between impeller and casings. Both these methods would reduce the skin friction, gyrostatic and axial losses. Credit is due the author for bringing these subjects before us and his studies of the subjects give us considerable information on these important losses, about which there is very meager and incomplete information.

M. D. HERSEY. In order to be able to state that an equation is wrong because it is not dimensionally homogeneous, it is necessary that all the physical quantities which govern that fact must be included in the question of the equation; hence, while the noteworthy use which the author has made of this criterion, both in refuting Rossiter's equation and in developing his own results, is entirely legitimate, it is to be remembered that, in other cases likely to arise in hydraulics, if some physical quantity, such as gravity, has been suppressed, an equation not dimensionally homogeneous may still be correct.

THE AUTHOR. When the surface of an impeller is covered with sediments equalling the structure of fine sand, the friction loss may increase to three times its value with polished surfaces, or twice that prevailing when the disk consists of a smooth casting. However, such degree of roughness may safely be termed abnormal in turbine pumps, and the loss caused by an ordinary smooth impeller in most pumps is not so much greater than that caused by a polished disk that it pays to polish the impeller faces, especially as they soon lose their polish when running.

As to the influence of eddying and lamellar motion, it should be emphasized that unless the speeds of the pumped fluid are high enough to insure eddying motion throughout, the ordinary centrifugal pump is not fit to pump. The papers presented by Messrs. Buckingham and Hersey at this meeting deal more closely with the problems touched by Mr. de Laval.

The suggestion to entrust a special commission in this country with the experimental work on viscosity is an excellent one. The author intends to direct the attention of the U. S. Bureau of Standards to the advisability of establishing more firmly the standards and laws of viscosity. He would be glad indeed to have Mr. de Laval coöperate in this motion.

Mr. de Laval very aptly draws attention to the effect which the heat developed by impeller friction must have on viscosity. The author, however, has not committed the mis-

take imputed to him—that of resorting to theoretical rather than experimental solution of the question of the influence of viscosity, temperature, and specific gravity of the fluid. As a matter of fact, he resorted to dimensional reasoning in this instance only after very carefully investigating all available experimental evidence.

After the paper was written, the author applied formula [23] to some other tests carried out with syrup of known viscosity, gravity and temperature, and found that the calculation tallied very well indeed with experiment. The figures are at Mr. de Laval's disposal.

Unfortunately the figures quoted by Mr. de Laval are not conclusive, as he does not state the value of the coefficient of viscosity of the oil which he pumped. This, by the way, is a figure which pump-makers are almost never able to obtain from customers, but without which they are entirely unable to predict results.

The idea of trying to rotate the impeller in air instead of water is not new. The author has himself given it a considerable amount of thought which he condensed in his application for German patent, No. N. 11099. There also are some French and American patents covering similar efforts. The practical difficulties which stand in the way of this idea, however, are very great indeed. How, for instance, is it to be avoided that the impeller should suck air from the lateral chambers?

At present those high lift impeller designs which tend to diminish the diameter of the impeller, i.e., to diminish the impeller coefficient, deserve more attention. The reduction in disk friction loss is about five times the percentie reduction of this coefficient.

The point raised by Mr. Hersey will be treated in my discussion of this gentleman's paper.

A BASIS FOR RATIONAL DESIGN OF HEAT TRANSFER APPARATUS

E. E. WILSON, NEW YORK

Non-Member[1]

AN examination of the literature on the subject of heat transfer shows such a wide variation in the coefficients established that it is difficult to make a choice for use in practice. The fact that the results of careful experimenters do not agree leads to the conclusion that some variable or variables have been neglected, and the fact that the rate of heat transfer in feed water heaters is consistently greater than that in the similar apparatus, the condenser, indicates that one such neglected variable is the temperature of the circulating water. It is the purpose of this paper to apply a correction for the water temperature to the results of reliable experiments.

By correcting for all variables, the rate of heat transfer which is generally expressed as some exponential function of the circulating water velocity, may be expressed in a different form as a straight line function. With this relation it is possible to evaluate the thickness of a water film capable of offering the same resistance to heat transfer as is encountered on the water side of a condenser or feed water heater. From the known internal conductivity of such a film it is possible to develop an expression for the area of heating surface required to transmit a given quantity of heat under all conditions of circulating water velocity and temperature, mean temperature difference, and tube diameters. With the use of suitable design factors this expression for area may form the basis of rational design, while a consideration of the manner in which the variables appear may be of assistance to the operator.

The possibility of solution along the lines indicated was brought to the author's attention by Dr. C. E. Lucke of Columbia University.

In condensers and feed water heaters the resistances to heat flow for clean tubes are:

a That due to the thickness of the tube walls.

b That due to the film of water on the steam side next the tubes.

c That due to air in the exhaust steam.

d That due to the water film on the water side.

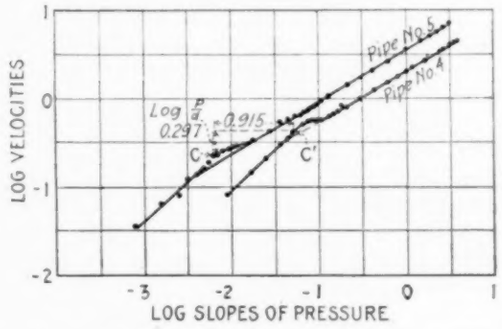

FIG. 1 VARIATION OF LOSS OF HEAD IN PIPES AS REPORTED BY OSBORNE REYNOLDS, PHILOSOPHICAL TRANSACTIONS OF THE ROYAL SOCIETY OF LONDON, 1883

Under ordinary conditions of operation (*d*) will be the controlling resistance and any increase in heat transfer should come through reduction of the film thickness on the water side. This is evident when the general law of heat transfer is considered. Taking the analogy of the electric or magnetic circuit we may write,

$$U = t_m \div R$$

in which

U = B.t.u. transmitted per hour per sq. ft.

t_m = mean temperature difference between the hot and cold fluids

R = resistance to transfer

Putting in the individual resistances enumerated above this becomes

$$U = \frac{t_m}{R_w + R_a + R_s + R_t}$$

in which

R_w = resistance of the water film on the water side

R_a = resistance of the air film on the steam side

R_s = resistance of the water film on the steam side

R_t = resistance of the tube walls.

If ρ be the resistance per unit of film thickness and l, the thickness of the film the expression becomes

$$U = \frac{t_m}{\rho_w l_w + \rho_a l_a + \rho_s l_s + \rho_t l_t}$$

[1] Lieutenant, U. S. N.

Presented at the Spring Meeting of the Society, June 1915. Complete paper may be obtained without discussion; price 10 cents to members; 20 cents to non-members.

in which we have the heat transfer in terms of the film thicknesses involved.

Considering now the controlling resistance, that on the circulating water side, we find that this decreases as the water velocity increases. Just how it decreases has been the subject of much investigation and the results have been summed up by G. A. Orrok[1] in his paper.

In his Scientific Papers, Osborne Reynolds pointed out that if a fluid traversed a tube without turbulence, the resistance to the transfer of heat would be entirely independent of the velocity; while, if the flow were turbulent, it would vary inversely as the velocity. In a later paper he showed that the transition from non-turbulent to turbulent flow was a discontinuous phenomenon: on a certain critical velocity being exceeded, turbulence abruptly sets in. Hence, when heat is being transferred from a hot tube to a fluid flowing in it, the law of heat transference changes abruptly once the critical velocity is reached.

The critical velocity according to Reynolds is given by the relation

$$V_{\text{critical}} = \frac{1}{B}\cdot\frac{P}{d}$$

where

P = Poiseuille's value for the ratio of viscosity to density

d = internal diameter of the tube.

B = a constant

The value of P in centigrade units is given as

$$\frac{1}{1 + 0.0336T + 0.000221T^2}$$

in which T is the absolute temperature centigrade.

If now the value of P at some one temperature, such as 60 deg., be settled upon as the standard, the ratio of P at any other temperature to that at the standard temperature may be computed as a specific viscosity.

The view that the rate of heat transfer should vary inversely with the viscosity was arrived at through consideration of the similarity between the accepted law for heat transfer and that for the loss of head in pipes. Reynolds found that up to the critical velocity the loss of head varied directly with the velocity as shown in the lower branches of two of his results in Fig. 1. The coördinates here are the logarithms of the loss of head and velocity, and the slope of the lower branches is 45 deg. At the critical points C and C' the flow becomes turbulent and the slope of the curve changes to about 1.72. Reynolds shows further that the temperature of the water and diameter of the tube have little or no effect upon the slope of the curves but that these may be shifted so as to be exactly coincident. The component of this shift in the direction of the velocity is 0.297 on the figure and this corresponds very closely to log. P/d in the expression for critical velocity above.

With these facts in mind the results for heat transfer were investigated and those reported by Orrok were selected as being the most complete and accurate. The mean temperature of the circulating water was computed from his factors of correction for viscosity and the data grouped in accord with certain average temperatures. The results are plotted in Fig. 2 from which it is seen that there is a distinct shift from the mean of the points for the lowest temperature to those of the highest temperature. Of course these results are too few and, not having been obtained with this purpose in mind, are too inaccurate for quantitative results, but they show well enough the qualitative effect.

The points defining the five lines of more or less equal temperature form a wide band. If, however, we apply a correction similar to that for the critical velocity so as to reduce the results to some common temperature, say 60 deg., the width of this band is cut in half. For instance, if the mean water temperature in one case were 40 deg. and the velocity 5 ft. per sec., the resistance to heat transfer would be the same as if the temperature were 60 deg. and the velocity

$$\frac{5 \times P_{60}}{P_{40}} = 3.62 \text{ ft-sec.}$$

This correction has been applied and the results as reduced from a 1-in. to a ⅝-in. pipe are plotted in Fig. 5 along with the uncorrected data. The reduction in the dispersion

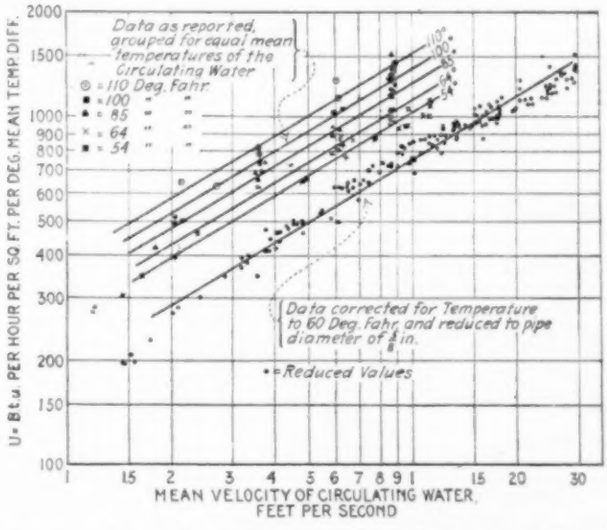

FIG. 2 RATE OF HEAT TRANSFER VERSUS CIRCULATING WATER VELOCITY. RESULTS OF TESTS BY ORROK, TRANS. AM. SOC. M. E., 1910

is quite marked and the agreement, in view of the difficulties in eliminating air from the steam in a test of this sort, becomes now quite good. There is a tendency toward curvature in the plots but this is not well enough defined to permit of revaluation owing to the scarcity of points at the low velocities.

By further reasoning it is found that there is a linear relation between the resistance on the water side and one variable, the velocity, after the other variables have been taken care of in reducing the velocity. The resistance of the tube walls for clean tubes is well known and that on the steam side has been determined for the one case, that of steam at 212 deg. The curves resulting from plotting experimental data will lie above that due to the sum of the resistance as calculated, because in practice steam is never quite air free and the resistance on the steam side is increased by that due to the air.

This constitutes the reduction for viscosity, but when the results of different experimenters are considered it is found that different sized tubes have been used.

But

$$\rho\omega = Rw \times \frac{\text{Inner diameter}}{\text{Outer diameter}}$$

[1] Trans. Am. Soc. M. E., Vol. 32, p. 1139.

$$(Rw)_1 = (Rw)_{\frac{5}{8}} \times \frac{1.00}{0.902} \times \frac{0.529}{0.625} \times \frac{0.529}{0.902} = 0.55\ (Rw)_{\frac{5}{8}}$$

gives a reduction factor for relating the two sizes of tubes on the basis of the resistance of the water.

As an example of the method of applying the corrections for working up the results, take the first run in Orrok's results. Here $V_w = 11.2$ ft. per sec.; $t_s = 126.3$ deg.; $t_m = 73.2$ deg.; $U = 1000$. Now $t_s - t_m = 54.1$ deg. From the table of correction factors, Table 1, $C = 1.10$. Then

$$V_{60} = \frac{11.2}{1.10} = 9.3$$

$$V_{final} = V_{60} \times \frac{0.902}{0.529} - 9.3 \times \frac{0.902}{0.529} = 15.9$$

of which the reciprocal is 0.063. This can be raised to the 0.82 power by laying off a pair of scales, one of which will show V when the other gives $V^{0.82}$. By this method the result is 0.102.

$$R = 1000 \times \frac{1}{V} = \frac{10{,}000}{1000}$$

or in this case, 1. The coördinates of a point on the final curve are then 1 and 0.102.

It is not absolutely necessary, as has been done so far, to deal with these resistances in terms of an arbitrary unit, since the resistance may be expressed in terms of the thickness of equivalent water films, by a simple transformation. When this is done we have the thickness of the films on each side and that of the tube wall expressed as a water film thickness, and we have the film thickness on the water side in terms of one variable which has been reduced in such a manner that all other variables are accounted for. In other words, we have a linear relation between the physical object offering the resistance and the factors upon which it depends for its magnitude.

It now remains to express the results algebraically and this may be readily done. Since the curves are of the form $y = mx + b$ we have, measuring the values of m and b on the line for "resistance as measured" in Fig. 3,

$$l = 0.0143 \left(\frac{1}{V_{60}}\right)^{0.82} + 0.00313$$

Now the reduced velocity V_{60} was corrected for both viscosity and the diameter of the standard $\frac{5}{8}$-in. tube, 0.529 in. If we let d be the inside diameter of the tube under test and Pt be the viscosity ÷ density for the mean water temperature as before and P_{60} the same ratio for the standard water temperature we can write

$$l = 0.0143 \left(\frac{P^t \times 0.529}{V \times P_{60} \times d}\right)^{0.82} + 0.00313$$

Calling the ratio of the viscosities C_w, water correction, and reducing we find

$$l = 0.0085 \left(\frac{C_w}{V \times d}\right)^{0.82} + 0.00313$$

If now C_w can be expressed in terms of the temperature we have an expression involving the film thickness and all the variables concerned. In Fig. 4 the values of C have been plotted against absolute temperature in deg. fahr. on logarithmic paper, and it is seen that while the curve is not exactly straight it can be divided into two sections which are practically so. The lower section has a range from 32 to 140 deg. fahr., which embraces the temperatures met in condenser practice and greatly exceeds the limits in this practice. Expressing the lower section algebraically we have

$$C_w = \left(\frac{520}{T_w}\right)^{0.57}$$

From which, and the expression for l above

$$l = \frac{34.0 \times 10^{12}}{T_w^{3.28} \times (V \times d)^{0.82}} + 0.00313$$

For water temperatures outside this range, such as those met in feed water heaters, the upper section may be used and a similar expression obtained.

This expression is a rational one as far as film thicknesses are concerned but is still not in a practical form. To reduce

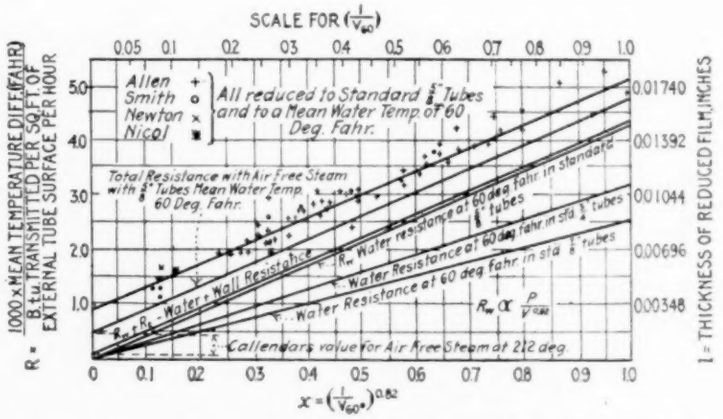
SCALE FOR $(\frac{1}{V_{60}})$
0.05 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8 0.9 1.0
$R = \frac{1000 \times \text{MEAN TEMPERATURE DIFF (FAHR)}}{\text{B.t.u. TRANSMITTED PER SQ FT OF EXTERNAL TUBE SURFACE PER HOUR}}$
5.0
4.0
3.0
2.0
1.0
Allen +
Smith o
Newton ×
Nicol ■
All reduced to Standard $\frac{5}{8}$" Tubes and to a Mean Water Temp. of 60 Deg. Fahr.
Total Resistance with Air Free Steam with $\frac{5}{8}$" Tubes Mean Water Temp. 60 Deg. Fahr.
R_w Water resistance at 60 deg fahr in standard $\frac{5}{8}$" tubes
R_w Water resistance at 60 deg fahr in std $\frac{1}{2}$" tubes
$R_w + R_s$ Water + Wall Resistance
R_w Water Resistance at 60 deg fahr in std $\frac{1}{2}$" tubes
R_w Water Resistance at 60 deg fahr in std $\frac{5}{8}$" tubes
$R_w \propto \frac{P}{V^{0.82}}$
Callendars value for Air Free Steam at 212 deg.
0 0.1 0.2 0.3 0.4 0.5 0.6 0.7 0.8 0.9 1.0
$x = (\frac{1}{V_{60}})^{0.82}$
0.01740
0.01392
0.01044
0.00696
0.00348
l = THICKNESS OF REDUCED FILM, INCHES

FIG. 3 RESISTANCE TO HEAT FLOW IN TERMS OF "FULLY REDUCED," VELOCITY OF CIRCULATING WATER. ENGINEERING, JAN. 23, 1914

it to such form, the general law of heat transfer is utilized. According to this law

$$Q = AUt_m$$

in which Q is the heat transferred per hour, A, the area of the heating surface, and t_m the mean temperature difference as before.

Remembering that we can now write

$$U = \frac{1}{\varepsilon l},$$

$$A = \frac{\varepsilon l}{t_m}$$

from which

$$A = \frac{Q}{3.48 \times t_m} \left(\frac{34 \times 10^{12}}{T^{3.28}(V \times d)^{0.82}} + 0.00313\right)$$

Upon further reduction this becomes

$$A = \frac{Q}{t_m} \left(\frac{976 \times 10^{10}}{T^{2.28}(V \times d)^{0.82}} + 0.0009\right)$$

This is a rational expression for the area of the heating surface required in a condenser in which all the surface is active and in which steam is substantially air free, as it exists when an effective dry vacuum pump is in use.

The above rational expression for the area of condensers, the determination of the actual thickness of the dead fluid film and its variation with the flow conditions, suggests a method of dealing with all heat transfer apparatus, dependent upon the same principles.

Conclusions. In condensers and feed water heaters, using

practically air free steam, the controlling resistance to heat transfer is on the water side. This resistance is a function of the water velocity, its mean temperature and the diameter of the tubes, and it varies inversely as these three variables, according to a straight line law. Large quantities of air collecting in pockets on the water side greatly increase the resistance to heat flow while the quantity of solid matter contained in the circulating water may increase the apparent resistance by increasing the apparent velocity as determined by the weight of water passing. The viscosity of the circulating water has an important influence on the resistance, since the warmer the water the less resistance it offers. The resistance due to oily circulating water may be greatly different from that of fresh water. Finally by analysis of reliable test data it is possible to get an expression for the area of heating surface required to transmit a given quantity of heat, in terms of prime variables, from which it is seen that the area varies directly as the quantity of heat to be transferred per hour per square foot, and inversely as the circulating water velocity and its mean temperature, as well as the tube diameter. Such an expression, based on good experimental results, should replace empiric coefficients in design and assist materially in operation.

DISCUSSION

ROBERT C. H. HECK stated in a written communication that the argument is developed so much by the experimental method that the title "rational" seemed to him to be fully deserved. The argument from analogy between the law for heat transfer and that for the law of head in pipes appeared to him to be rather far-fetched, the functions used not being really enough alike in their manner of variation to warrant such an assumption. Besides, the assumption that general velocity states are proportional to critical states is not yet proved. He could not accept the assumption that because, with the same water temperature, the critical velocity in a 1-in. tube is only one-half as great as in a half-inch tube, therefore the unit surface of the larger tube is twice as effective as the unit of the smaller tube. The idea of complete inverse proportionality to temperature appeared to be too simple and not yet proved experimentally. He believed that the difference of slant in the final line of Fig. 7 and 8 was due more to excessive influence of tube diameters than to the causes named by the author.

LEO LOEB objected to the fundamental conclusion of the writer that the control resistance to heat transfer from the condensing vapor to liquid warming rests on the liquid side and that the numerical value depends on the three factors of water velocity, mean temperature and tube temperature. Recently types of apparatus have been produced in which the agitation of the liquid is carried to the highest practicable point consistent with reasonable frictional resistance and it appears that in the near future the largest part of such apparatus will embody construction principles which will make it altogether impossible to base results on either velocity or temperature.

Variation in results reported by several investigators may be due to lack of standard methods of tests and of uniformity in the presentation of data as well as, in some instances, to an improper conception of the meaning and application of the term "temperature difference." Thus accidental errors may be due to difference in temperature across the cross section of a pipe where the outlet temperature is measured. Several instances have shown that water is discharged from different portions of the heating surface in streams of varying temperature and that these streams did not diffuse readily.

He objected to the assumption generally made that the law of heat transfer is analogous to that of an electric or magnetic circuit which it can be only when there is under consideration a single thermal resistance, a case which never occurs in practice.

He further discussed his own tests conducted at the Naval Engineering Experiment Station upon single tubes and normal heaters (Published in the Journal of the American Society of Naval Engineers, May, 1915; compare abstract in The A.S.M.E. Journal, August, 1915), and their bearing on the theory presented by Lieutenant Wilson.

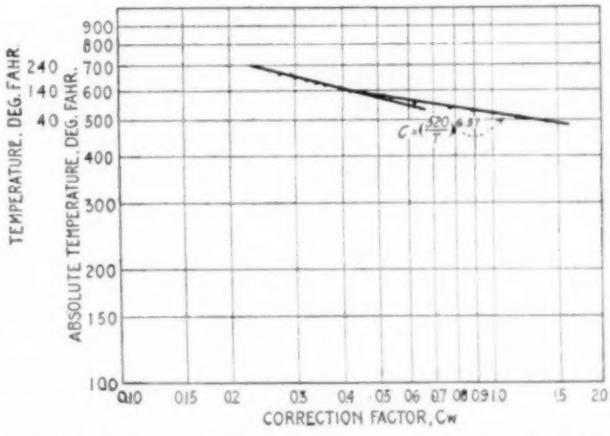

FIG. 4 CURVES OF CORRECTING FACTORS FOR VARIATION IN VISCOSITY DUE TO TEMPERATURE OF CONDENSER CIRCULATING WATER

H. WADE HIBBARD made a plea that whenever mathematical papers are presented before the Society the authors use footnotes showing how they have evolved their equations, in the most elementary fashion. He had observed that mathematical papers presented before the Society in years past had received scant discussion, due in part to the jumps in the mathematics in the papers.

The author mentions that "The controlling resistance to heat transfer is on the water side," showing how much emphasis he places upon the word "controlling." In condenser practice we have rain plates so that the moisture will be removed from the tubes as quickly as possible and the steam which follows will strike against tubes that are not encased in wet film.

EDGAR BUCKINGHAM contributed a written discussion of the physics of the subject of heat transmission, in which he showed that the heat transmitted to the liquid from unit length of the tube in unit time could be expressed by an equation of the form

$$Q = DV\rho C\Delta\Phi\left\{\frac{DV\rho}{\mu}, \frac{V^2\alpha}{C}, \frac{\mu C}{\lambda}, \alpha\Delta\right\}$$

where

D = internal diameter of tube
V = speed of fluid
ρ = density of fluid
μ = viscosity of fluid
C = specific heat of fluid
Δ = temperature difference
λ = conductivity at mean temperature of section
α = temperature coefficient of conductivity between this and temperature of inner surface of tube
Φ = an unknown function which remains to be determined.

This equation shows that a complete investigation of the problem of heat transmission, even for a single liquid in so simple apparatus as a condenser tube, is rather complex. To determine the value of Φ it is necessary to find the effect on Q of varying four arguments of Φ separately. In view of our almost complete ignorance about the thermal conductivity of water, to say nothing of other liquids, it is not surprising that we do not yet know all there is to know about heat transmission; and any attempt to proceed rationally, as Lieutenant Wilson has done, by taking into account and allowing for the various physical quantities that may be involved in the process of heat transmission is very much to be welcomed.

ARTHUR M. GREENE, JR. I would like to ask the author how he proposes to get the mean temperature so as to use the results of his work, and also how he got the mean temperature which he did use.

As Mr. Braun points out, if he uses the arithmetic mean or the logarithmic mean, he gets two results; and if the viscosity is affected by the temperature he has got to consider that fact in finding the coefficient of heat transfer. It seems that both of those methods in getting t_m are incorrect and you have really to find t_m by the method that Orrok uses in his work on the condenser.

I would also like to say that Mr. Leo Loeb's paper, which recently appeared in the Transactions of the Institute of Naval Engineers, clearly points out the fact that this heat transfer coefficient varies with the temperature. The method he uses is rather unique, and proved clearly, I think, that we must use some exponent form for the computation of T_m. If we do this it seems to me better than to use an expression involving viscosity.

C. F. BRAUN. This paper is particularly interesting to the writer because it treats in a most thorough manner a factor in calorifier design, which is given only slight attention in my condenser paper.

Unquestionably the resistance to heat flow on the fluid side of a tube increases rapidly with increase in viscosity and is a most important factor in the design of calorifiers for such viscous fluids as oils, syrups, etc., having viscosities ranging up to 50 times that of water. As the oil industry particularly is an important one in this country and the demand for oil heaters and exchangers large, the development of rational formulae relating to heat transfer and viscosity would be very valuable, and can the author extend his researches to these fields he will render a service to the profession.

The viscosity of water varies so slightly within the ranges of temperatures encountered in heater and condenser practice that its effect, I feel, is almost negligible compared with that of other variables indeterminate and difficult to control. I am a great believer in figuring, but we must eliminate unimportant complications, or we avoid and do not use our theory.

One of the greatest variables in determining the unit coefficient of heat transfer for comparison of different experimental results is the method of computing the mean temperature difference. Without discussing the merits of the various formulae, it is sufficient to call attention to the fact that the arithmetic mean, logarithmic mean and others produce widely varying results, particularly when the ratio between the least temperature difference and the greatest temperature difference is small.

Again the amount of air present is a variable which cannot be controlled or its effect determined, and will invariably produce variations far beyond the limits of viscosity effects.

The density or mass flow of the steam, while apparently having no important effect upon heat transfer, may very probably have as much if not more effect than the viscosity of the water.

I also thoroughly believe that heat transfer varies inversely with tube size and to a far greater degree than with water viscosity. Experimental data on this point would be valuable.

Unaccountable discrepancies in condenser test results are very frequently the result of assuming a uniform temperature in the steam space, instead of obtaining an average which may fall considerably below that at the inlet, due to frictional pressure drop through the tube of space, and to the presence of air.

THE AUTHOR. Professor Heck criticises the reference to the similarity between the laws of resistance to the flow of water and heat on the grounds of dissimilarity of exponents. The mere numerical value of these exponents is of little importance for the purpose in hand. It was the similarity in form of the accepted law that led to the study, and although this study showed the exponential law to be incorrect for heat transfer, nevertheless the remarkable facility with which all data are reconciled by the viscosity correction is justification enough for the method.

Professor Heck also objects to the use of the word "rational." My understanding of the meaning of this word as applied to an equation is that an expression is rational when it so involves the different variables that the truth may be checked up by the fundamental dimensional equations. Doctor Buckingham's very interesting paper treats of that same subject. Taking the expression for film thickness in the paper we have,

$$L = \frac{a}{V} \div b$$

where L is the thickness, V, the reduced water velocity, a, the slope of the line, and b, the intercept on the vertical axis, Figs. 7 and 8. In this expression b has the dimension L; V is of course L/T; a is L/V or L^2/T, and we then have

$$L = \frac{L^2T}{TL} \div L = L$$

which is a rational expression. Referring to the expression for the area we have

$\frac{A = Q \not{t} L}{t_m}$ in which $L/$ is taken as the expression above.

by definition $Q/t_m = U$, and $P = L/U$ from which

$$A = \frac{UL^2}{U} = L^2$$

which expression is again rational.

As for the influence of the tube diameter to which Professor Heck takes objection, there seems no reason for endeavoring to find a more complicated function. The resistance to heat flow and water flow are both a function of the water agitation. In the flow of water the diameter is shown by Reynolds to have a certain influence on the critical point, that is the point at which the flow becomes turbulent, and this feature in turn affects the resistance to flow. It seems not unreasonable to expect the diameter to have the same sort of influence on heat transfer, which is dependent upon the agitation in the same manner. In comparing a half inch and a one inch tube as to the effectiveness of the unit surface it must be borne in mind that for the same linear velocity the quantity of water passed increases as the square of the diameter. If now the flow is turbulent, there can be a larger number of impacts on the unit surface of the larger tube than of the smaller, and this accounts for the greater effectiveness.

Professor Greene has asked the method of obtaining the mean water temperature. This is shown in the tables where I have subtracted the mean temperature difference t_m from the temperature of the steam t_s, thus getting t_w, the mean water temperature. Both Mr. Braun and Professor Greene have discussed the mean temperature difference. I am aware of the difference of opinion as to the proper way of getting this. Mr. Orrok in his calculations used the arithmetic mean for the reason that the difference in results by different methods when the temperature rise is small is not worth considering.

Mr. Loeb states that no useful purpose can be served by considering the high circulating water temperature in design where space and weight are limited. On the other hand this seems to me to be the very place careful design is needed. As an example take the condenser design of a battle cruiser developing say 120,000 horse power, on a small displacement. Her condensers will be beyond the limits of present practice so that no comparison may be had with other vessels, yet, on the other hand, space will be so confined as to preclude guess work. Surely accurate design is needed here.

Mr. Loeb calls attention to the fact that the variation in results reported by several investigators may be due to experimental errors, rather than the neglect of the variables mentioned. I appreciate fully the difficulties of work in this field and drew Fig. 9 as an example. My point is that the wide variations in the results of so excellent an investigator as Mr. Orrok are not explained by experimental errors but by neglected variables instead. When in Fig. 5 of the paper I corrected his results for temperature, they became reconciled to a point where further difference might be assigned to experimental inaccuracies.

The objection by Mr. Loeb that the analogy of the law of the electric circuit can apply only when we have a single resistance is not sustained. We are fully justified in writing such a law if we remember that the resistance in this case is a variable. Mr. Loeb practically uses the same analogy in his paper, but instead of separating the resistance into its component parts he assumes that he is keeping this constant when he maintains the water velocity constant and determines an exponent for the mean temperature difference. In the general expression,

$$U = \frac{t_m}{R_w \div R_s \div R_a \div R_t}$$

R_w is not dependent upon the velocity alone, but also on the mean water temperature, and his failure to consider the temperature is probably responsible for the value of his exponent for t_m, and its difference from unity. Any exponential law for heat transfer involving either V or t_m, however convenient it may be for the design of a limited class of apparatus, cannot be considered a valid general law. As Mr. Loeb says any one can run a few tests and use them as a basis for design for similar apparatus involving the same conditions. Surely, however, we are not entitled to use these few results in writing a general law for the whole subject. Mr. Loeb, in his paper, has developed in a novel manner an excellent method of designing certain types of apparatus, but these same data could not be extended to other types.

The objection that new types of apparatus have been evolved in which agitation is carried to the limit so that viscosity need not be considered, is hardly valid. We have two distinct methods of reducing the resistance, mechanical agitation and increased water temperatures. In certain types of apparatus the first method is carried to the limit, but this does not remove the influence of temperature on the thickness of the water film.

In regard to the value of the exponent of V in the paper, there is no connection between this exponent and that used for t_m by Mr. Loeb in his article. The value 0.82 was settled on by the author of Fig. 7 in *Engineering* apparently by trial. I have since plotted the results of Mr. Orrok's work against V with the index unity and find excellent accord. Further experimental work must determine the real value of the exponent.

In conclusion I want to point out that successful design of heat apparatus has been done empirically and will continue to be so done. In all design, however, we must appreciate two great classes, (1) Functional Design and (2) the Design of the Mechanism. Mr. Braun in his paper gives an excellent summary of the design of the mechanism of a condenser. My paper is intended to demonstrate a basis for rational functional design of heat transfer apparatus of all types for every flow condition, throughout the whole range of variation.

FOREIGN REVIEW AND REVIEW OF PROCEEDINGS OF ENGINEERING SOCIETIES

ENGINEERING SURVEY

In the section Air Engineering, attention is called to the article by Prof. Brabbée and Dr. Bradtke on the graphical process for the determination of dimensions of pipes in ventilating and heating systems, developed on the basis of an extensive series of tests. In the same section is reported the beginning of an article on graphical tables for calculating reciprocating compressors. Both articles are interesting as showing a strong tendency toward development of graphical methods for solving engineering problems.

The article by Binder on initial temperatures of combustible gases and liquids is an interesting contribution towards a question of considerable interest to the designers of gas and explosion engines (though it is quite possible that the data indicated therein have been already discounted empirically).

In the section Mechanics, is reported an article on recent tests of the effective forces between driving belt and pulley. Among other things, these tests have shown the fallacy of the idea that the belt delivers a higher frictional resistance in gliding on a rough pulley than on a smooth one.

In the same section is continued from the August Journal the abstract on theory of resistance to rolling of a hard body over a plastic surface.

From a German paper is abstracted an account of experience with coke firing under boilers, indicating some limitations of this kind of fuel, and giving suggestions as to the best methods and most suitable type of equipment.

A discussion of the location of the neutral zone in heated buildings is presented from a paper before the American Society of Heating and Ventilating Engineers.

F. L. Fairbanks before the American Society of Refrigerating Engineers describes the design and installation of a large ammonia compression machine, giving in detail the system adopted for ordering the machinery and its installation. The suction valve of the refrigerating machine is described and illustrated, and some indicator diagrams are given from the ammonia end of the compressor.

The fundamental problems of engine design and formulae for comparison of gasoline and automobile performances are reported from two papers presented before the Society of Automobile Engineers.

From the Journal of the South African Institution of Engineers is taken an abstract of a paper on the influence moisture in air has on mine ventilation. Among other things, the author considers the physiological effect of excessive moisture and gives some rules as to the basic methods of designing mine ventilation systems.

FOREIGN REVIEW

Air Engineering

SIMPLIFIED GRAPHICAL OR ANALYTICAL PROCESS FOR THE DETERMINATION OF DIMENSIONS OF PIPES IN VENTILATING AND HEATING INSTALLATIONS, Professor Brabbée and Dr. Bradtke.

Abstract of No. 21 of *Communications of the Testing Laboratory for Heating and Ventilation,* of the Royal Technical High School, Berlin (the original communication is not available in the Library of the Engineering Societies). It is devoted to the calculation of high pressure and steam heating and ventilating installations by a method previously established for the determination of the diameters of piping in hot water heating systems.

The following notation is used:

H = Total pressure in kg/m²
R = Frictional resistance per meter of piping in kg/m²
Z = Fall of pressure due to single resistances in kg/m²
L = Volume of air in m³/sec.
f = Cross section of the passage in m²
l = Length of passage in meters
v = Velocity of air in m/sec.
d = Diameter of passage in millimeters
g = Acceleration due to gravity in m/sec.²
γ = Specific weight of air in kg/m³
a = A constant
ξ = Resistance coefficient.

In the equation

$$H \geq \sum_{1}^{n} (lR + Z) \quad [1]$$

all frictional resistance of the air may be expressed by a potential function having the following form:

$$R = \frac{a v^n}{d^m} \quad [2]$$

The single resistances are taken care of by the following equation:

$$Z = \xi \frac{v^2}{2g} \cdot \gamma \quad [3]$$

In connection with these three equations, the following is added:

$$L = fv \quad [4]$$

Equation [2] has been derived from the data of a considerable number of investigations reported in technical literature on frictional resistance of air in metal passages. It is based on 293 observations made on passages having a diameter from 19 to 1000 mm., and since a remains constant only for a definite specific weight of air, the corresponding specific weight γ had to be brought in accordance with the conditions existing in the ventilating passages. This was done by means of data on the influence of γ on the air resistance derived from the very careful investigations of Fritsche. Further were used data obtained experimentally by Rietschel on the flow of air in rectangular sheet iron passages. There were no modern investigations covering the case of brick passages, but it was found that the values obtained from equation [2], when doubled, were in nearly perfect accordance with the values given for brick passages in Rietschel's *Handbook for the Calculation and Design of Ventilating and Heating Plants* (in German). Therefore, the final formula in which the average roughness of the ventilating passages is taken care of by varying the velocity exponent n can be applied both for the round and rectangular sheet iron and brick passages.

Among other things, the investigation gives a table indicating the values which have to be used in the above equa-

tions. There are four more tables covering the following subjects: 1, equivalent diameters and cross-sections of passages for rectangular passages from 0.1 to 2.5 m. length of side; 2, effective pressures in passages 1 m. high, prevailing at various external and internal temperatures; 3, single resistances generally occurring in practice; 4, calibration constants for frictional resistances at γ equals 1.2 and temperatures in the passage varying from 0 to 80 deg. The new method of calculation is illustrated by three examples. (*Vereinfachtes zeichnerisches oder rechnerisches Verfahren zur Bestimmung der Rohrleitungen von Lüftungs-und Luftheizungsanlagen,* Professor Dr. Brabbée and Dr. Bradtke, *Gesundheits-Ingenieur,* vol. 38, no. 28, p. 325, July 10, 1915, 2 pp., *p.*)

GRAPHICAL TABLES FOR CALCULATING RECIPROCATING COMPRESSORS, Immerschitt

M. Hirsch ("Air Pumps, the Design, Calculation and Testing of Compressors and Vacuum Pumps," in German) was the first to offer a complete method for graphical calculation of reciprocating compressors. His method, however, did not take into consideration losses in the suction opening of the pressure valves and expulsion of the compressed air from the cylinder. The reexpansion of the air from the clearance space is also taken into account.

When such diagrams are plotted, certain relations have to be taken into consideration. First, compressors equipped with automatic valves are here considered primarily. Further, assumptions have to be made as to the character of the air (whether dry or moist), as to the cooling exponent of the compression and re-expansion line, and as to the dimensions of the clearance of the compressor. In plotting the suction and expulsion lines the speed of the compressor has to be also considered. Hence the diagram does not apply to any kind of compressors, but only to those equipped with automatic valves and jacket cooling, and not to slide valve compressors, compressors with the equalitation of pressure, semi-wet and wet compressors. This does not, however, matter much because semi-wet and wet are scarcely built any more, and compressors with automatic valves and jacket cooling are mostly used.

As regards the selection of data of design the clearance space was taken to be 2.5 per cent. Fig. A shows the air diagram for 7 atmospheres absolute air compression com-

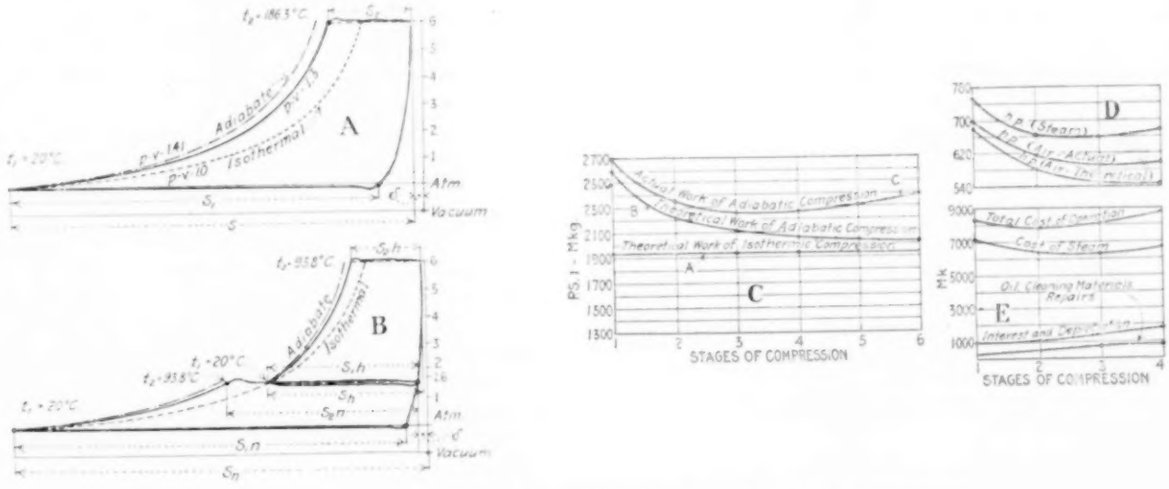

FIG. 1 DIAGRAMS SHOWING COMPARATIVE PERFORMANCES OF SINGLE AND MULTISTAGE COMPRESSORS

of air and in its displacement from the cylinder due to the motions of the valves and valve resistances. As a result, in order that the data graphically obtained by the Hirsch method might be used, they had to be multiplied by certain coefficients obtained experimentally.

The present author attempts to offer a graphical method for handling reciprocating compressor problems based on somewhat different principles. In order to obtain tabular values for the single stage and two stage compression and for different end pressures, compressor diagrams were plotted in shape very near those found in actual practice. Fig. 1A shows a diagram for single stage compression and 6 atmospheres gage pressure; and Fig. B a diagram for a two-space compression with the same end compression. In these diagrams plotted from data found in numerous diagrams taken in actual practice, all the resistances met with in the operation of a compressor with automatic suction and pressure valves, are found to be taken into account, such as losses in opening of the suction valve, taking in of the air, in compression because of the heating of the air, pressors. In construction of the compression and re-expansion lines the exponent 1.3 was adopted which pre-supposes an effective cooling, without, however, making serious demands on it. The suction pressure is taken to be 5 per cent. below the outside air pressure, while the compression pressure (the use of light automatic valves being assumed) is 0.15 atmospheres higher than in the compression chamber. Multi-stage compression results in lower power consumption of the compressor for the same air output, greater volumetric efficieney, better maintenance of the compressor and drier compressed air. As shown in Figs. C, D and E the most favorable from a technical point degree of compression is attained on the assumption of 7 kg. per sq. em. end compression, in three stages. In four-stage compression the losses through friction in suction and additional compression in the fourth stage are greater than the gain obtained through the triple return cooling as compared with the double one. If in addition the increased work in piston friction in three stage compression is taken into consideration as well as the additional cost of the second intermediary

cooler, and the higher consumption of cooling water, it is found that the most desirable reciprocating compressor for compression from 6 to 7 atmospheres is the two-stage one.

Hence the tables worked out apply for single stage compression to air pressures of 3-8 atmospheres absolute, and for two-stage compression from 5-20 atmospheres absolute; while the three-stage compression has been left entirely out. Fig. D gives an air diagram for two-stage compression at 7 kg. absolute air pressure.

The diagrams have been plotted in such a way that the work done in low pressure and high pressure cylinders and the end temperatures be equal to one another. Hence the compression ratios in both of them must be equal or both must be equal to the square root of the absolute end compressor. (*Graphische Tabellen zur Berechnung von Kolbenkompressoren,* Immerschitt, *Der praktische Maschinen-Konstrukteur,* vol. 48, nos. 27/28, July 15, 1915, article not finished; abstract will be continued in an early issue.)

Internal-Combustion Engineering

INITIAL TEMPERATURES OF COMBUSTIBLE GASES AND LIQUIDS

For the proper operation of gas, benzole and gasoline engines, the explosion temperatures generated by the combustion of the fuels are of great importance. There is scarcely any doubt that the temperatures as previously computed on the assumption of the specific heats being constant were too low. Therefore the author has recomputed these temperatures, using increasing specific heats, and the difference in results, as shown by Table 1, are quite large.

In accordance with the more modern data on specific heats increasing with temperature, it is found, for example, that the specific heat of 1 cbm of carbon dioxide at 1000 deg. cent. is as follows:

$$1000(0.4886 + 0.00024 \times 1000)$$

Hence the increase of temperature for 1 deg. cent. is equal to $0.4886 + 0.00024 \times t$.

And from this is computed the general formula for the combustion and explosion temperatures:

$$T = \frac{WE}{cbmCO_2(0.4886 + 0.00024t) + cbmH_2O(0.4692 + 0.00015t)} \frac{WE}{cbmN(0.308 + 0.00007t)}$$

The initial temperatures have been computed with theoretical air volumes and on the basis of combustion with pure oxygen; for explosives such mixtures may perhaps find their application. The temperatures when taken with pure oxygen are at rising temperatures.

In the table the volumes of the explosive mixtures are given before and after combustion, and from this are computed the gage pressures available during explosion. Of considerable interest are the usually small differences in the initial temperatures, which fact the author explains by assuming that what occurs is the conversion of oxygen into carbon dioxide and water vapor. Since, however, the difference in heating value of these two elements in equal contact with oxygen is not particularly great, the initial temperatures also cannot vary very much. The extremely powerful noise occurring during explosion of acetylene is, therefore, due less to the development of a high pressure than to some sort of acoustic action.

Sooner or later these much lower initial or explosive temperatures will have to be taken into consideration in designing gas and explosion engines, unless practice has already discounted these temperatures empirically. (*Ueber Anfangstemperaturen von brennbaren Gasen und Flüssigkeiten,* O. Binder, *Oel- und Gasmaschine,* vol. 15, no. 4, p. 25, July 1915, 2 pp., *et.*)

Mechanics

RECENT TESTS ON THE EFFECTIVE FORCE BETWEEN DRIVING BELT AND PULLEY

The law of friction of solid bodies applies fully only to the case of surfaces of belts and pulleys clean and absolutely free of grease and the nearest approximation is the case of new belts having very little grease on them. If on the other hand there clings, more or less perfectly, on the smooth surfaces of the belt and pulleys a thin skin of liquid, then, in accordance with the amount of its adhesion, the magnitude of the effective force varies as the internal friction of the liquid and thus becomes functionally dependent on all other variables, in particular: the effective surface, the gliding velocity, and the temperature and viscosity of the adhering liquid. Under such conditions new forces come into operation, in some cases many times greater than those acting in the case of pure friction between solid bodies. The properties of the belt material become of secondary importance, while the properties and amount of belt grease assume a preeminent importance.

The presence of a uniformly thin and smooth skin of grease on the gliding surface of a belt has a double effect. In the first place, it makes possible the rise of large forces between belt and pulley, especially with increase in gliding velocity. Second, it protects the surface of the belt. In particular for high belt velocities belts should be as flexible, soft and well greased as possible. In the case of a slightly greased belt, the application of a proper belt grease to the clean surface can help in building up a thin skin of grease between the belt and the smooth pulley which raises the effective forces as has been proved by experiments.

When there is a skin of liquid present, the magnitude of the effective forces increases, in the first place, with the gliding velocity. This makes the belt drive stable as regards overloads (up to certain definite limits of this latter); when large peripheral forces have to be transferred, higher gliding velocities must be used, and they lead to increased frictional resistances on the assumption that the temperature remains permanently constant. The average gliding velocity of the belt and pulley increases (all other conditions being the same and peripheral forces transferred being equal) approximately in proportion to the belt speed. In the high speed belts, therefore, usually larger frictional forces are in operation.

The superiority of large pulley diameters and comparatively wide belts, established by experiments, is partly explained by the functional dependence existing, in the case of well greased belts, between the magnitude of the effective forces and that of the gliding surfaces.

With rough surfaces of pulleys the effective force is greater than with the smooth pulleys only when the velocity of gliding is negligibly small. Otherwise it is always smaller, the more so the more perfectly the face of the belt is covered by a thin skin of fluid. As a result rough pulleys not only cut down the life of the belt through increased

wear but do not accomplish the purpose of increasing the frictional resistance in gliding. (*Versuche über die Grösse der wirksamen Kraft zwischen Triebriemen uns Scheibe*, A. Friedrich, *Zeits. des Vereines deutscher Ingenieure*, vol. 59, nos. 27, 29 and 30, pp. 537, 580 and 608, July 3, 17 and 24, 16 pp., 39 figs. *et.A.*)

Theory of Resistance to Rolling of a Hard Body over a Plastic Surface, B. B. Schultz

A treatment of the resistance to rolling of cylindrical wheels over various types of ground, considering in particular resistance to rolling of driven wheels. The present article, while an independent unit, is, to a certain extent, a further development of an article previously published by the author and abstracted in *The Journal* August 1915, p. 478.

As regards the resistance to rolling of driven wheels, the author gives in a table the various coefficients (determined experimentally) of resistance of motion of ordinary wagons over horizontal roads of various kinds, compiled from data found by Morin, Résal and Watson. None of these investigations indicates how the resistance to motion varies with the change of diameter of the driven wheels, the width of their rims and presence or absence of gripping protuberances, and what are the most favorable dimensions of a wheel giving minimum resistance to rolling. In order to answer these questions, the author breaks up the complex phenomenon of rolling into its simplest components, by considering three of its elements: viz. *first*, resistance to motion due to friction in the journals of the wheels of the wagon; *second*, due to vibrations of the body of the wagon, and *third*, resistance of ground to rolling of the rims of the wheels.

The resistance due to friction in the journals of the wheels of the wagon may be considered as a braking force R_o kg applied horizontally to the axis of the wheel, and, mechanically, may be expressed by the following formula,—

$$R_o = \mu \frac{d}{D} \cdot Q$$

where d is the diameter of the journal of the shaft, D the diameter of the driven wheel, Q the load on the shaft in kilograms and μ coefficient of friction on the surface of the journal. According to M. Forestier, this coefficient is 0.1 for ordinary journals lubricated with journal grease, 0.01 for patent bearings lubricated with oil, and 0.005 to 0.0025 for lubricated ball-bearings. According to the investigation made by the author, the diameter d of the journal does not exceed, in the case of heavy trucks, 60-80-100 mm, and for light wagons 40-50 mm, so that the ratio $\frac{d}{D}$ does not usually reach 0.110 and does not fall below $\frac{1}{30}$, being on the average approximately $\frac{1}{20}$. On the whole, the author shows that resistance to the motion of the wheel, due to friction in the shaft, does not exceed usually one-half of one per cent. of the load of the shaft and in the least favorable case, does not reach one per cent. of that load.

The resistance to motion due to vibration of the wagon is considered as the entire increase of resistance to motion due to the velocity of motion. The author comes to the following conclusions as to this particular item in the resistance to motion: The variation of the resistance to motion of a wagon with the increase of its velocity is observed only when motion occurs over uneven hard ground where this resistance is great, and in the the first approximation, the resistance increases with the velocity, in a straight line. Resistance to motion due to the vibration of the wagons in the case of motion over uneven, hard ground, at very low velocities, amounts to not more than some tenths of one per cent of the load of the wagon. In rolling over smooth ground, the resistance to rolling of hard tires may be considered as practically independent of the velocity. The resistance to the motion of the wagon over soft ground (subject to deformation), is on the average greater than the resistance to motion over every other kind (hard) of ground, and reaches the maximum absolute value (up to 25 per cent) of the load on the wheels. The resistance to rolling of wheels over soft ground (subject to deformation), is the main constituent of resistance to motion.

The state of theory as regards resistance to rolling is rather unsettled. According to the law of friction in motion formulated first by Coulomb and accepted by Morin, this resistance is proportional to load and inversely proportional to the diameter of the rolling cylinder; according to Professor Hele-Shaw, it is inversely proportional to the square root of the diameter; and according to conclusions drawn from the present investigation, it is inversely proportional to the diameter raised to the power 2/3. Further, the coefficient of resistance to rolling is, according to Coulomb and Morin, independent of the load, while according to tests

TABLE 1. INITIAL TEMPERATURES OF EXPLOSION OF COMBUSTIBLE GASES AND LIQUIDS

	Initial Volume with oxygen	Final Volume with oxygen	Initial Volume with theoretical amount of air	Final Volume with theoretical amount of air	INITIAL TEMPERATURES at constant specific heat		INITIAL TEMPERATURES with varying (upwards) specific heat		PRESSURES	
					with oxygen	with theoretical amt. of air	with oxygen	with theoretical amt. of air	with oxygen	with theoretical amt. of air
	C. B. M.	C. B. M.	C. B. M.	C. B. M.	°C	°C	°C	°C	Atmospheres	
Hydrogen	3,00	22,93	6,771	42,01	6781	2756	2851	1711	7,63	6,205
Carbon Monoxide	3,00	21,775	6,771	43,361	7067	3042	2694	1775	7,26	6,53
Methane	3,00	34,15	10,542	71,78	7160	2440	2829	1583	11,38	6,86
Acetylene	7,00	85,49	25,855	203,06	....	....	3610	1953	12,21	7,85
Ethylene	4,00	51,34	15,313	114,093	8620	2750	3226	1758	12,84	7,45
Benzole	17,00	231,65	73,565	544,47	....	2790	3234	1717	13,63	7,40
Gasoline	12,00	183,90	53,481	400,78	....	....	3068	1661	15,33	7,50
Alcohol	4,00	55,06	15,313	110,063	....	....	2728	1566	13,77	7,19
Illuminating Gas	2,135	22,95	6,435	45,51	7500	2530	3126	1747	10,75	7,07

made by the committee of the British Association, it decreases with the load. According to tests at the automobile laboratory of Professor Riedler, it increases with the load, and according to the data of the present investigation, it increases with the load slightly and is proportional to the cube root of the load.

The present investigation is limited to the consideration of resistance to rolling of driven wheels moving over a horizontal and plastic roadway. The specific resistance of the ground to crushing is assumed to be proportional to depth y of crushing, so that $f = f_1.y$, where f_1 in kg per cbm is the coefficient of specific resistance of the ground to crushing.

If we denote by Q kg the load on the surface of rolling, by R kg the effort of rolling applied to the axis and forcing the wheel to roll uniformly, by B cm the width of the rim and by y_o cm the depth of the impression left in the ground by the rolling cylinder, then the specific work L_o in kg per cm required to deform 1 sq. cm of the surface of the ground to a depth y_o, will be in kg-cm

$$L_o = \int_o^{y_o} f.dy = \int_o^{y_o} f_1.y dy = f_1 \frac{y_o^2}{2}$$

and hence the work per unit of length of roadway (1 cm) or the magnitude of the resistance to rolling will be

$$R = \frac{L}{l} = L_o.B. \quad \ldots\ldots\ldots\ldots [1]$$

Now (Fig. 2 A) let the origin of coordinates be at O; the axis of X along the line OA; the axis of Y along the line OB and the axis of Z normally to the plane of the drawing. If we take an element $dS = B.dl'$ on the surface of the rolling cylinder, which dl' is the element MN of the arc AB; let $dF = f.dS$ be an element of the force of pressure of the ground against the surface of rolling corresponding to dS; assume f the specific pressure on the ground to be proportional to the depth of crushing y : $f = f_1$. If we neglect friction, we by that very fact assume that $dF \perp dS$. External forces applied to a uniformly rolling body, must be in equilibrium, and therefore the algebraic sum of their vertical projections must be equal to zero: $\Sigma Y = 0$ or

$$Q - \mathrm{S}dF.\cos\alpha = 0$$

where $\mathrm{S}dF.\cos\alpha$ must be spread over the entire surface S of rolling in contact with the ground. But $dF.\cos\alpha = B.f.dx$, since $dF.\cos\alpha = (F.ds)\cos\alpha = FBdl'.\cos\alpha$; on the other hand, as one can see from Fig. 2 B, $dl'.\cos\alpha = dx$, and hence

$$dF.\cos\alpha = F.B.dl'.\cos\alpha = F_1.y.B.dx \quad \ldots\ldots [2]$$

which brings the equation to the form:

$$Q = F_1.B.\int_o^{y_o} y dx \quad \ldots\ldots\ldots\ldots [3]$$

The properties of the circle indicate the following (Fig. 2 C):

$$x^2 = [D - (y_o - y)](y_o - y) \quad \ldots\ldots [4]$$

If we differentiate this equation, free it from x and substitute in equation [3], we have then:

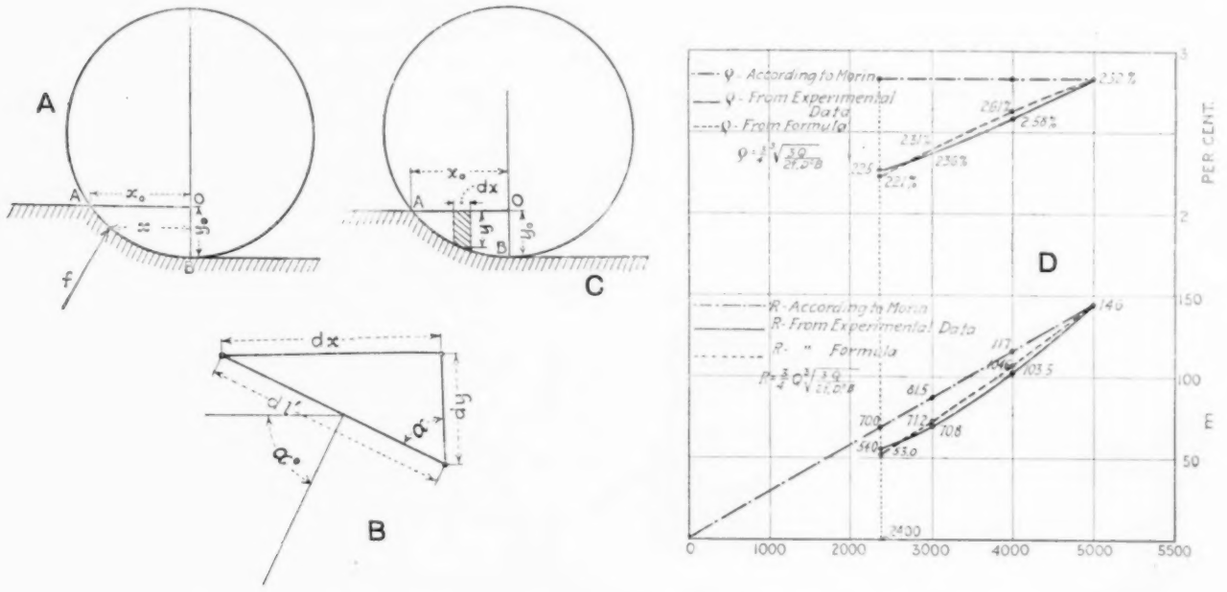

FIG. 2 THEORETICAL CURVES OF RESISTANCE TO ROLLING AS COMPARED WITH THOSE OBTAINED EXPERIMENTALLY

$$Q = F_1.B\int_o^{y_o} y \frac{D - 2(y_o - y)}{2\sqrt{[D-(y_o-y)](y_o-y)}} \quad \ldots [5]$$

where the fact has been already taken care of that, in differentiation, the signs of ydx and dy are opposite.

It is theoretically possible to solve the complicated integral in equation [5], but practically there would not be much use in doing so. By approximate substitution, the author derives the following two expressions (the numbers of the equations are those in the original article):

$$Q = \frac{2}{3}.F_1.B.\sqrt{D}.y_o^{3/2} \quad \ldots\ldots [7]$$

$$\varphi = \frac{3}{4}\sqrt[3]{\frac{3Q}{2F_1.D^2.B}} \quad \ldots\ldots [12]$$

where $\varphi = \frac{R}{Q}$, and is the coefficient of resistance to rolling.

Formulae [1], [7] and [12] give the solution of the problem of resistance to rolling in the case of a cylindrical wheel with a hard rim, moving over a plastic roadway, on the assumption that the resistance of the ground is proportional to the

depth of its deformation. The author shows that the expression for R obtained in equation [1] by means of computing the work of deformation of the ground, may also be obtained from the equations stating the conditions of equilibrium of a rolling wheel, by equating to zero the algebraic sum of the horizontal projections of all forces acting externally on the wheel. From this, he derives the following equations, which give a general and analytical solution of the problem of resistance to rolling of a driven wheel with a hard cylindrical rim, moving along a plastic roadway, this solution being independent of the law of resistance of the ground to deformation:

$$R = B\int_0^{y_o} F.\mathrm{d}y \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [13]$$

$$Q = B\int_0^{x_o} F.\mathrm{d}x \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [14]$$

Theoretically, therefore, the problem can always be solved, no matter what it is, by the experimental law of resistance of the ground to the deformation. Equations [13] and [14] can be integrated with particular ease in the case when f = constant = f_o; i.e., when the specific resistance of the ground to deformation is constant, in which case

$$R = F_1.\mathrm{B}.y_o = \frac{\mathrm{Q}^2}{F_o.B.D} \ldots\ldots\ldots\ldots [15]$$

$$\varrho = \sqrt{\frac{y_o}{D}} = \frac{Q}{F_o.B.D} \ldots\ldots\ldots\ldots [16]$$

$$Q = F_o.B.\sqrt{D}.y_o \ldots\ldots\ldots\ldots [17]$$

$$y_o = \frac{\mathrm{Q}^2}{F_o^2.B^2.\mathrm{D}} \ldots\ldots\ldots\ldots [18]$$

It is, however, important to bear in mind that, as shown by the author, the law of resistance assumed above has been shown, by the experiments of the author, not to be in accordance with actual conditions. From this, the author proceeds to show the magnitude of error caused by the use of the simplified expression for x^2 in equation [4].

There is no way of easily verifying the correctness of the formulae derived by the author, although theoretically a single typical test, properly carried out, would have been sufficient for this. The equation for $\varrho = \frac{3}{4}\sqrt{\frac{y_o}{D}}$ derived on the assumption of the resistance of the ground and the depth of its deformation being proportional to one another, foretells the magnitude of the coefficient ϱ, which is, on the other hand, equal to the ratio of experimental magnitudes, R/Q. If, now, we write in a general form

$$\varrho = \xi\sqrt{\frac{y_o}{D}}$$

we can show by means of the initial equations that ξ depends on the law of resistance of the ground to deformation and, in particular, if the resistance is constant (independent of depth), then

$$\xi = 1, \text{ and } \varrho = \sqrt{\frac{y_o}{\mathrm{D}}}$$

From this is obtained a remarkable conclusion, viz., that if the coefficient ξ be determined by means of data obtained experimentally from the equation

$$\frac{R}{Q} = \xi\sqrt{\frac{y_o}{D}} \ldots\ldots\ldots\ldots [19]$$

it is possible from its magnitude to judge of the nature of the law of deformation of the ground in the expectation that the smaller ξ is as compared with unity, the more rapidly will the resistance of the ground increase with the depth of deformation.

From equation [1] for R, it follows that the absolute magnitude of the resistance to rolling of a cylindrical wheel over a given ground is proportional to the width of the rolling rim B and the square of depth y_o of the imprint of rolling, provided the resistance of the ground is proportional to the depth of deformation. From another equation, it follows that if the resistance of the ground is proportional to the depth of deformation, the specific resistance to rolling of the wheel $\varrho = \frac{R}{Q}$ is deteremined entirely by the specific magnitude of the penetration of the wheel into the ground $\frac{y_o}{D}$. From the same formula, it follows that a relatively deep penetration of the wheel into the ground is a necessary and sufficient index of a large relative resistance to rolling. From equation [12], it follows that for a given wheel and roadway the relative resistance to rolling is not a constant magnitude but one slightly increasing with the load (proportional to $Q^{\frac{1}{4}}$) if the resistance of the ground is proportional to the depth of its deformation. It may be noticed in this connection that experiments in the laboratory of Professor Riedler have shown the practical independence of resistance to rolling of a truck, from the velocity of rolling over a given ground, and also the increase of the relative resistance to motion with the load.

Diagram 2 D gives a comparison of theoretical curves of resistance to rolling with curves obtained experimentally, at the Riedler laboratory, on tests with a truck equipped with iron tires and rolling over wooden boards. This diagram shows, *first*, that the independence of the relative resistance to rolling assumed by Coulomb and Morin in their general law of resistance to rolling, is not true in this particular case since it disagrees with the experimental data to the extent of about 30 per cent; *second*, likewise in the given case is not correct the assumption that the specific resistance of the ground to deformation is constant, since it disagrees with the experimental data to the extent of about 40 per cent. The formula of resistance to rolling of a hard cylinder over a plastic roadway derived above on the assumption of the resistance of the ground being proportional to the depth of deformation satisfied in a manner sufficient for practical purposes the qualitative data obtained from experiments.

Equation [12] shows also that the magnitude of the relative resistance to rolling of a hard wheel over a given plastic roadway increases (slowly) with the load per unit of width of the rim of the wheel, in proportion to $\frac{Q^{\frac{1}{4}}}{B}$ and decreases more rapidly with increase of the diameter of the wheel (inversely proportional to $\mathrm{D}^{\frac{3}{4}}$). A 3 per cent increase of the diameter of the wheel decreases the resistance to rolling by 2 per cent, while an increase of the width of the rim of the wheel by 3 per cent decreases the resistance to rolling by 1 per cent, so that in order to reduce the resistance to rolling, it is advisable to use wheels of larger diameter even though of smaller width. The depth of deformation of the ground decreases slowly with the increase in the diameter of the driven wheels, and more rapidly with the reduction of load from unity of weight on the rim of the wheel; hence, in the interest of the preservation of roads, wide wheels, even though of small diameter, should be used. The author

recommends, therefore, in the interest of road preservation, that the load per unit of width of rim should be regulated by legislation, in accordance with the formula which he likewise suggests.

Railway Engineering

EQUALIZATION OF MASSES IN MOTOR DRIVEN LOCOMOTIVES AND OSCILLATIONS DUE TO IT, H. Henich

Discussion of the problem of equalization of masses, or rather, lack of equalization and oscillations due to it on motor driven locomotives. Because of lack of space only a very brief abstract of this interesting article can here be given.

In motor driven locomotives the driving plant consists usually of a horizontal slow-speed single cylinder engine; the reciprocating parts of a driving engine create centrifugal forces directed horizontally while the engine is running. They, in their turn, create forces directed vertically and varying with time as a function of their magnitude, in addition to which the centrifugal forces of the second order are free to act.

Because a locomotive is located elastically on its underframe, there arise inside the locomotive, on account of the action of the vertical forces, oscillations which are limited only to motions up and down if the driving engine is located so that its axis is in the plane of the centre of gravity of the locomotive. These oscillations in their turn create not inconsiderable additional forces which strain the axes and springs of the locomotives, and put an additional load on the roadway.

The author develops the formulæ for the calculation of oscillatory displacements, and of the additional forces acting therein. It is further shown that there is a "critical ratio" between the speed of the rotation of the engine and weight of the locomotive, and the constant of the spring, at which the amplitude of oscillation and the additional forces corresponding to it are theoretically "infinite" and practically "very large." Next it is shown what is the influence of a change in the locomotive springs or the speed of rotation of the engine on the magnitude of the amplitude of oscillations or the additional forces.

The author discusses also the movements due to centrifugal forces of the second order and shows what may be expected in the growth of the oscillatory phenomena if the driving engine be located not in the plane of the centre of gravity of the locomotive; but if, instead, multi-cylinder horizontal engines be used as a driving plant. In conclusion it is shown what type of engines are best to be used in larger sizes of locomotives, whenever a quiet running of the locomotive is considered of really serious importance. (*Zur Frage des Massenausgleiches und der dadurch enstehenden Schwingungen bei Motorlokomotiven*, H. Henich, *Der Oelmotor*, vol. 4, nos. 1 and 3, pp. 3 and 102, April and June 1915, 10 pp., 5 figs. *dt.*)

Steam Engineering

COKE AS A FUEL UNDER BOILERS, G. Wirthwein

Discussion of the utilization of various kinds of coke as boiler fuel. Both blast furnace and gas cokes are considered.

The average analysis of blast furnace coke is given as follows:

	Per Cent.
Moisture	8.88
Ash	9.81
Carbon	78.02
Hydrogen	0.54
Sulphur	1.03
Oxygen	0.86
Nitrogen	0.86
	100.00

The moisture content of coke varies in accordance with its origin. This variation is sometimes ascribed to moisture taken in by the coke during transportation, but experiments by Dr. Thaler show that this may be so only to a limited extent. Table 2 shows the amount of moisture absorbed by coke after lying submerged in water for corresponding periods varying from one to twenty-eight hours. From these tests it appears that while sample No. 1 has absorbed 10.2 per cent, increase in the other samples was as low as 2.2 per cent. (No. 4) which was probably due to the fact that sample No. 1 was very porous and sample 4 very hard. Dr. Thaler concludes from his tests that in transportation of porous coke even during strong rains, increase of moisture content will not exceed 4 per cent., and with hard coke from 1 to 2 per cent.

TABLE 2 ABSORPTION OF MOISTURE BY COKE KEPT UNDER WATER (WATER CONTENTS IN PER CENT)

No. of Sample	Original Water Content	Moisture absorbed after coke has been kept under water for Hours:				
		1	3	6	24	28
1	8	18.2	17	17	15.7	14.4
2	8.5	14.5	13.3	13.3	13.3	13.3
3	6.7	11.2	11.2	....	10.1	8.9
4	10.4	12.6	12.6	....	11.5	11.5

The experience of German factories and electric plants with coke firing since the beginning of the war was approximately as follows: It was found that coke alone could not be burned at all on traveling or chain grates. That might have been expected. When coal is burned on this type of grate the ignition of the gases brings the arch to a very high temperature, and as a result coals even very rich in gases can be burned on such grates practically without generation of any smoke. On the other hand, when coke is burned there is no flame at all and as a result the arch remains at a lower temperature. Hence when the coke is moved forward on the grate, it does not find a zone of incandescence, which leads to the fire simply going out. On the other hand, a mixture of three-fourths coal and one-fourth coke can be burned quite successfully on such grates.

The experiments with burning coke on the ordinary flat grate proved far more encouraging as it was found that even pure coke could be burned when the conditions of draft were favorable. This was especially easy to attain with artificial draft such as is produced by a fan or steam jet blower; for example on locomotives even pure coke could be easily burned in large lumps, presumably on ac-

count of the sharp draft there available. In the district of the Bergisch Steam Inspection Association several important concerns have adopted hand firing on flat grates, using a mixture of two parts of coal to one part of coke and without encountering any serious difficulties.

When the coal-coke mixture is fired on the horizontal grate, the fireman must look out for certain particulars. The furnace must be fired up with pure coal of as good a quality as is available; the coal-coke mixture is not used until after the fire has burned up well and is of sufficiently high temperature; and the fuel bed has to be kept somewhat higher than with coal. For a time before cleaning the clinker it is advisable to use unmixed material, that is pure coal, so that the fire shall burn away uniformly, as only in this way are the complaints prevented that unconsumed coke is taken out from the fuel bed with the clinker.

On the whole it was found that the gas coke is a more suitable material for boiler furnaces than blast furnace coke. Large sized coke can be burned on flat grates with hand firing only when a strong draft and a good grate cooling are available, as for example, in locomotives. For firing small coke and coke breeze not mixed with coal, special grate types with undergrate blowers are required. This type of coke can be burned on ordinary flat grates only when mixed with an easily baking coal. In general, it appears however that there is quite a wide field in steam boiler firing where coke can be usefully employed without large and expensive alterations in furnace design being required. (*Koks und seine Verwendung für Dampfkesselfeuerung*, G. Wirthwein, *Zeits für Dampfkessel und Maschinenbetrieb*, vol. 38, no. 29, p. 241, July 16, 1915, 3 pp., 1 fig. *egp.*)

Miscellanea

Recent Tests on Flow of Water Acted on by a Propeller

Description of tests made in the small experimental basin of the Royal Technical High School in Berlin. The outbreak of the war has caused a discontinuance of the tests, but some of the tests have been found to be sufficiently advanced to make it advisable to publish their results. Pitot tubes were used as measuring instruments throughout.

The article describes the experimental arrangements and methods of carrying out the tests and reports in a general manner the results obtained, given in more detail in the form of numerous curves. One of the interesting phenomena observed is the fact that behind the propeller in the region of the hub a very powerful flow of water *towards* the propeller was found. Continual variations of the water column as high as 170-180 mm have been observed. In the region of the periphery of the hub turbulent motions were found and only near the vanes was the water found to flow backwards. This was the water which was sucked in by the propeller, accelerated in it, thrown back by its motion, and in this way made effective for driving the ship forward.

The greatest velocities of water were found at low speeds within a region extending from 0.25 to 0.75 of the radius of the propeller. At higher speeds of rotation the maximum of water velocity shifts more toward the end of the vanes. Thus at 800 r.p.m., it lies at about 0.6 of the radius all around the propeller; from 0.75 of the radius both the pressure and the velocity curves at nearly all speeds investigated fall off constantly and at about 0.82 radius pass through the zero line. But beyond 0.82 of the radius there is again found the flow of water towards the propeller. These various velocities of flow are of different strengths at various distances from the axis of the shaft. Thus, for example, at 0.92 of the radius the water column gave the same indication as at 0.98 of the radius. This indicates the presence of eddies at the edges, occurring in addition to the proper propeller flow. Only at a distance of 1.25 radius is there again found quiet water all along the circumference.

All this data has been obtained by measurement over a cross section located about 20 mm behind the rear edge of the propeller blade. Essentially therefore they apply only to this cross section, but since the shape of the "screw water" is nearly cylindrical, these figures apply approximately also to several other cross sections behind the propeller. Fig. 3 gives a diagramatic view of a possible occurrence of water currents both in front and behind the propeller. Only components in the longitudinal direction are therein represented, so that the values in the direction of maximum flow are not determined and the rotation of the water is not indicated (figure is not drawn to scale).

As regards the flow of water towards the screw in the region of the propeller line, the author believes that it is

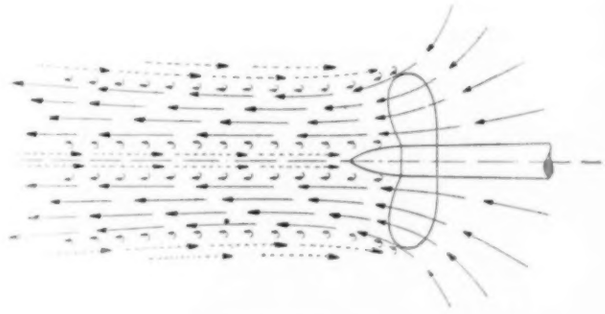

Fig. 3 Image of Flow of Water Displaced by a Propeller

permissible to assume that the partial vacuum behind the hub materially affects the efficiency of the propeller. It appears therefore desirable to make the diameter of the hub as small as possible, that is, as small as the strength of material and the method of fastening the propeller blades make it possible. In front of the propeller, that is, in the water under suction, the stream lines are quite different from those in the water under pressure behind the propeller. In the first place the water column in U-tubes in front of the propeller does not rise as high as in the other case; the flow in the opposite direction in the region of the hub cross-section is entirely missing. The influence of the propeller extends over a very much larger cross section; for example, at 7.5 mm in front of the forward edge of the propeller vane at higher speeds of rotation, quite a noticeable amount of water would be disturbed at 1.5 radii.

The rotation of the water filament has been observed here also, just as in previous tests where such rotation was noticed in sand and air. It differs, however, from the rotation seen behind the propeller through the fact that several cylinders of rotation are formed simultaneously. (*Neuere Untersuchungen und Messungen im Schraubenwasser mittels Düsen*, Professor Flamm, *Schiffbau*, vol. 16, no. 20, p. 553, July 28, 1915, 13 pp., 18 figs. *et A.*)

ENGINEERING SOCIETIES

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

Vol. 21, no. 2, July 1915, New York City.

The Determination of Pipe Sizes for Hot Water Heating Systems, Professor F. E. Giesecke (Part 2)
Can We Locate the Neutral Zone in Heated Buildings? J. J. Blackmore (abstracted)
Engineering Data for Designing Furnace Heating Systems, Prof. A. C. Willard
The Establishment of a Standard for Transmission Losses from Buildings of all Constructions, R. P. Bolton
The Establishment of Standard Methods of Proportioning Direct Radiation and Standard Sizes of Steam and Return Mains, James A. Donelly

CAN WE LOCATE THE NEUTRAL ZONE IN HEATED BUILDINGS? J. J. Blackmore

Discussion of the location of the neutral zone in heated buildings; derivation of formulae for the determination of its location, and discussion and consideration of the effect on the neutral zone of the shape of a building.

A neutral zone is a place or a plane inside a room or

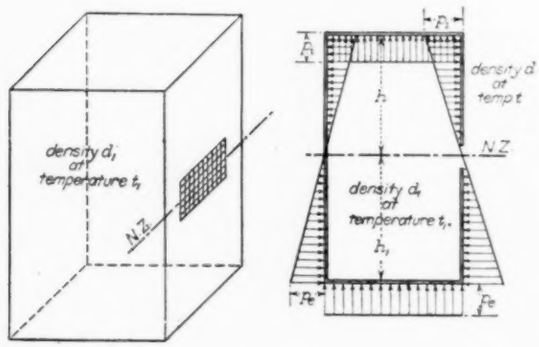

FIG. 4 NEUTRAL ZONE IN HEATED BUILDINGS

building where the pressure is equal to that of the air surrounding it on the outside. Its exact location in the room or building depends upon the relative leakages at the ceiling or upper part of the wall or at the floor or lower part of the walls, and under ordinary conditions in still air, its location will be slightly below the central horizontal plane of the room.

In cold weather, the air contained in a heated building is much lighter than the air surrounding it and therefore has a tendency to rise in accordance with the laws of gravitation. Since it is kept from rising by the walls and roof of the building, a pressure is exerted against the ceiling and upper part of the walls. The distribution of the pressure on the walls and ceiling depends, as the author shows, on the location of the openings connecting the inside of the room with the outside cooler atmosphere. Thus, in Fig. 4, with an opening in the side of the room, the neutral zone will be slightly above the center of the opening, and the following formula will apply:

$Pi = h_1 + h(d - d_1)$ to obtain the pressure by density.

$$Pi = h_1 + h\left(\frac{0.0864 \times 460}{460 + t}\right) - \left(\frac{0.0864 \times 460}{460 + t_1}\right)$$

to obtain (the pressure by temperature) where Pi is pressure above the atmospheric; d density of outside air; d_1 density of inside air.

The author derives similar formulae for the case of a zone located at a point below the floor line (this condition being produced by the floor register and the ventilating pipe with a strong draft), as well as for a reverse condition with a ventilating pipe from the ceiling. The above formulae apply to the cases of rooms supposed to be airtight and uniformly heated to 70 deg. when the outside air is at zero.

The author discusses briefly how the location of the neutral zone affects the heating conditions and ventilating system of a building. Its location determines the amount of radiation necessary in the various parts of the building to maintain the uniform temperature on the inside, as well as the ventilating pressures in the building. (11 pp., 15 figs., *gt.*)

AMERICAN SOCIETY OF REFRIGERATING ENGINEERS

Journal, vol. 2, no. 1, July 1915, New York City

Design, Construction and Operation of a 1000-ton Ammonia Compression Refrigerating Machine, F. L. Fairbanks
Decomposition of Ammonia
Condensation of Gasoline from Natural Gas
Boiler Losses
A. S. M. E. Boiler Code

Extraction of Humidity from Buildings by Refrigeration.

DESIGN, CONSTRUCTION AND OPERATION OF A 1000-TON AMMONIA COMPRESSION REFRIGERATING MACHINE, F. L. Fairbanks

Description of the design and installation of a large ammonia compression refrigerating machine, of particular interest because it describes in detail the system adopted for ordering the machinery and for its installation.

The Quincy Market Cold Storage and Warehouse Company of Boston, Mass., had, some time ago, to install a large new unit on a very limited floor space, and the installation had to be made in a short time and under very rigid specifications. No builder could be found in a position to handle the work in the specified time and according to the required standard, and it was decided to design and erect a machine with the company's own men contracting for the individual parts with several of the largest machine, foundry and forge shops.

The contracts were based upon a price per pound for each detailed machine part as per drawing and upon the estimated weights of the designer, and paid on the basis of actual weights of castings as delivered to shop to be machined; on difficult machine work, the prices remained fixed regardless of weight. The company furnished drawings, patterns, foundations, cranes, rigging and men for the erection. The drawings carried the specifications for each different material and, where necessary, the physical and chemical characteristics. All important measurements, standing and running fits, and clearances for expansion as well as running clearances, were shown on the drawings in thousandths of an inch, thus making it possible to have multiple parts machined by different men and in separate shops without reference to each other, and at the same time have them fit together accurately when assembled, as proved to be the case.

The designer was to check, test, and accept or reject each individual piece. The patterns were checked before and after molding. The molds were checked after the cores were in and before closing, and the castings were carefully exam-

ined to detect defects before they were machined, all with a view to finding and rectifying errors before they had developed in time and money. This proved an efficient method as there was no part of the machine rejected for defect either in material or workmanship after its arrival at the power plant for erection. The immense pieces to be installed were handled by the company's own men, partly because of the fact that practically all the material had to be carried on cranes over running engines; although the rigging was carefully tested, when parts were passing over some of the larger compressors, a man was stationed at every important steam and ammonia valve with orders to shut it off if he heard a carsh: everything went on, however, without any trouble.

The article describes the installation in considerable detail. The apparatus is mainly of standard design, but special attention has been paid to the design of the suction valve (Fig. 5 D) as this feature alone, unless very carefully worked out, will nullify practically all of the benefits to be derived from such an installation. The admission of the high pressure gas from the piston ports is in the nature of an explosion and this at a time when the suction valve is wide open,

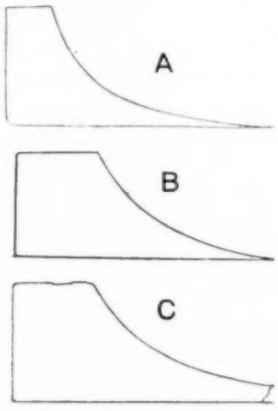

FIG. 5 A, B AND C, INDICATOR DIAGRAMS FROM AMMONIA END OF THE REFRIGERATING MACHINE

resulting in a very forcible closing of these valves. Even with large area, low lift, good lubrication and quick action, in a design worked out for known local conditions, smoothness of operation begins to disappear at between 35 and 40 r.p.m. Speeds above 50 r.p.m. are attended with more or less valve slamming, rapidly increasing discharge temperatures and consumption of power.

Fig. 5 A, B, and C are reproductions of diagrams from the ammonia end of the compressor. Fig. A shows the compressor cylinders operating with low pressure gas; Fig. B shows the same with gas from the high-pressure system and Fig. C shows the machine operating with the combined gases. When the gases were combined, the piston ports, as shown by the indicator card, are also large enough to allow of the filling of the cylinder at speeds up to 50 r.p.m. before the piston has reached the end of its stroke. To the very small clearance is probably due the fact that there is practically no re-expansion of gases, it being possible to produce practically square corners on a card by feeding a few extra drops of oil to the cylinders per minute. (29 pp., 24 figs. *d.*)

THE FRANKLIN INSTITUTE

Journal, vol. 180, no. 2, August 1915, Philadelphia, Pa.

The Effects of the End Connections on the Distribution of Stress in Certain Tension Members, Cyril Batho (abstracted)

Modern Theories of Magnetism, G. F. Stradling

Standardized Colored Fluids, H. V. Arny and C. H. Ring

An Investigation of Fusible Tin Boiler Plugs (U. S. Bureau of Standards)

The Testing of Rubber Goods (U. S. Bureau of Standards)

THE EFFECTS OF END CONNECTIONS ON THE DISTRIBUTION OF STRESS IN CERTAIN TENSION MEMBERS, Cyril Batho

Investigation of the distribution of stresses in single or built-up structural members and its modification due to different types of end connections.

The paper describes in detail the very interesting method

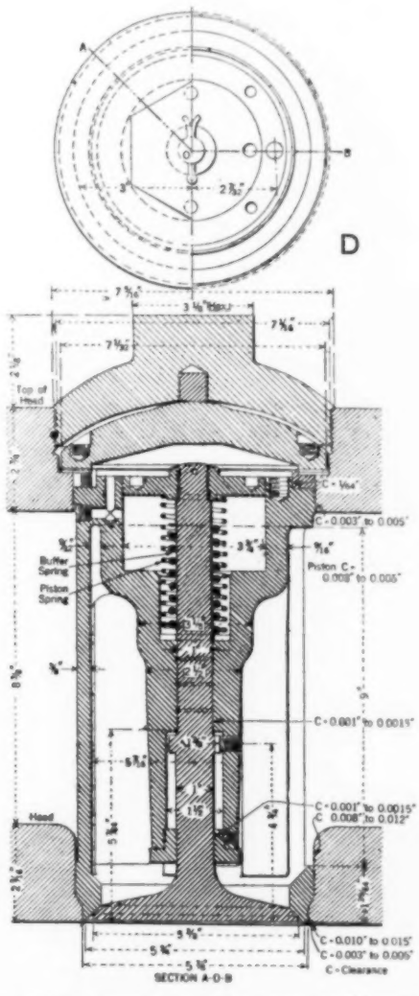

FIG. 5 D SUCTION VALVE, REFRIGERATING MACHINE

of investigation applied. The peculiarity of the latter lies in the use of a simplified form of Martens' mirror extensometer, constructed in the laboratories of the McGill University, Montreal. This instrument, while very simple in construction and operation, was shown to be capable, when certain precautions are observed in its use, of measuring strains accurately to 1/100,000 in. on a length as small as

2 in., and of being used in the most confined positions, such, for example, as the space between two angles placed back to back and separated by as little as ⅜ in.

The paper further reports in detail Prof. L. J. Johnson's method of determination of distribution of strains, and among other things, his S-polygon, a figure which gives at a glance the point of maximum bending stress and the value of the latter for any given load axis. The author arrives at the following conclusions:

First. The only practicable experimental method at present available for investigating the distribution of stress in built-up members is by means of some form of extensometer, and the simplified mirror extensometer used in the tests described is very suitable for this purpose.

Second. The assumption of a planar distribution of stress is justifiable in such members as are considered here, except perhaps close to the end connections, and the ordinary theory may therefore be applied to an analysis of the distribution of stress in these members.

Third. In single- and double-angle tension members connected at their ends by means of rivets to wide and rigidly held gusset plates, the stiffness of the gusset plate in its own plane has a considerable effect on the distribution of stress in the member, there being in every case a particular stiffness which will give the least maximum stress in the member for a given load.

Fourth. In such members lock angles are of very little, if any, value for the purpose either of giving a more equable distribution of stress in the member or of increasing the effective length of end connections.

Fifth. A slight change in the line of application of the load to the gusset plates does not materially affect the distribution of stress in the member, except possibly close to the end connections.

Sixth. The experiments on double angles bear out the theory that such members do not act as a single piece bending as a beam (44 pp., 22 figs., *et A*).

SOCIETY OF AUTOMOBILE ENGINEERS

Bulletin, vol. 8, no. 4, July 1915, New York City

Fundamental Problems of Engine Design, A. P. Brush (abstracted)

A Formula for the Comparison of Gasoline Automobile Performance, Cornelius T. Myers (abstracted)

Motor Vehicle Governors, Theodore Douglas

FUNDAMENTAL PROBLEMS OF ENGINE DESIGN, A. P. Brush

Discussion of the fundamental problems of engine design, mainly with regard to what would be the best engine for a modern automobile.

The author, together with the majority of automobile engineers, assumes that the small-bore, long-stroke, high-speed motor is the best adapted to automobile service, at least pleasure vehicles. This is because it satisfies the two fundamental requirements,—low weight and flexibility. The latter has to satisfy demands through a range unapproached in any other service since if 3 to 60 m.p.h. be accepted as reasonable minimum to maximum road speed on top gear, there is a 95 per cent speed fluctuation and all through this speed range the motors must operate satisfactorily from zero to full power.

As regards the most advisable length of stroke, the author points out that to meet the demands for flexibility, a power curve must be secured that will give good values above 2500 r.p.m. This gives what might be termed the valve-chamber-limit on stroke, because with an assumed form of cylinder and combustion chamber, the valves are limited to an outside diameter slightly less than one-half of the bore of the cylinder and to a throat diameter further reduced by the width of the valve seat.

As regards the connecting rod length, the time-factor as effecting combustion chamber efficiency requires a short connecting rod. The time-factor is most prominent during the latter portion of the compression and the earlier portion of the working stroke. Due to its angularity, the shorter the rod, the less the time proportion of the compression and working strokes coincident with high pressures, high temperatures and high wall-to-volume ratios. Therefore, the shorter the connecting rod the higher the combustion chamber efficiency at any given number of revolutions per minute.

As regards the mooted question of the number of cylinders, the author points out that the problem of increasing the power and flexibility of an automobile on top gear is solved, with the least reduction in efficiency, by increasing the number instead of the size of the cylinders, besides which the frequent impulse engine is more pleasing to the driver in its operation throughout the entire speed range; but there is a price to be paid for it, as there is an increase in mechanical complications. In particular regard to V-motors, the author calls attention to the fact that if all cylinders were worked properly on a single crank throw, all major vertical vibratory tendencies present in the vertical motor would disappear and in their place would be found horizontal vibratory tendencies which are twice as great as the vertical vibratory tendencies of the vertical motor. In the V-12 there are the same major vibratory tendencies as in the vertical 6; that is, instead of a tendency to reciprocate the entire motor mass with four reversals per revolution of the crankshaft, as in the vertical four and V-8, there is a tendency to set up periodic vibrations within the motor mass, the amplitude of these vibrations being in proportion to the work the engine is doing.

In the discussion which followed, C. W. McKinley stated that he had run some tests of two engines of practically the same volumetric capacity, one of which had an almost spherical combustion chamber and the other one pancake shaped which had practically as large a surface as it was possible to make, allowing proper gas flow over the valves. With both of these motors exactly the same horsepower was secured and the speaker therefore believes that the shape of the combustion chamber has very little effect on the ultimate power of the motor.

Victor W. Pagé called attention to the fact that the multiple cylinder engines take very much more time to repair and that for example in valve grinding, the probable expense would be $2.00 to $3.50 to grind in eight valves on a four; $4.00 to $7.00 for the valves on an eight and $6.00 to $10.50 for the valves on a twelve. He believes therefore that the car owners will feel it also after having paid several "twin-multiple" repair bills.

David Furguson spoke of a test recently conducted by the Pierce-Arrow Company. One of its 2-ton trucks using a standard 4-cylinder 4 by 5½ in. engine with a standard worm gear having ratio of 7 to 1, did, on a long run, 7.25 miles per gallon of gasoline. A similar machine with an imported high speed engine of one-half the cylinder volume, geared 14 to 1, was run alongside the standard machine and

did only 6.6 miles per gallon. The speaker asked, therefore, if high speed multiple cylinder engines are more efficient than medium speed 4-cylinder engines, why are the six, eight and twelve high-speed engines not used on trucks, in the case of which economy is of so much moment?

D. McCall White, in speaking of the number of cylinders, urged the development of the small high-speed four, and strongly deprecated the stampeding (of which there is evidence) by some companies who manufacture the lower-priced class of automobiles, into the building of eights. An eight must be of the highest and most expensive type of manufacture. It is easier and cheaper to build a good high-speed four than a bad high-speed eight, and a good four will be more satisfactory to all concerned than a cheap eight. As regards the sixes and twelves, the speaker is of the opinion that, given the same rigidity of bearings on cam shaft and crank shaft, their mechanical efficiency is not so good as that of the four and eight.

Howard E. Coffin stated that the requisite number of cylinders depends entirely on the size of the car. The question is not whether a four, six, twelve or sixteen cylinder engine should be built, but how large the individual cylinder shall be. If the piston displacement is over 300 cu. in. it can properly be broken up into a greater number of cylinders than six, but where the total piston displacement is less than 300 cu. in., it is foolish to break it up into more than six cylinders.

Howard Marmon stated that he had never seen an eight or twelve do things that a six will not do, except possibly some of the "stunts." Speaking of vibrations, he called attention to the fact that the eight cylinder transforms the purely vertical vibration into a horizontal one. The vertical vibration is taken up by the springs, which have a slow period. The chassis is rigid laterally. Any rigid structure will transmit high frequency vibrations as the cross vibrations will necessarily be, and the speaker's opinion was that the cross vibration of the eight was very much more pronounced than the vertical vibration of the four (25 pp., *g*).

A Formula for the Comparison of Gasoline Automobile Performance, Theodore Douglas

Discussion of a formula which would make it possible to compare various motor vehicles. It gives the following expression for the vehicle coefficient:

$$\text{(A)} \qquad VC = \frac{8.4nb^2sR}{DW}$$

where n = number of cylinders; b = bore in inches; s = stroke in inches; R = gear reduction; D = diameter of driving tires in inches; W = total weight of vehicle and load in pounds.

In addition to that, VC is multiplied by e_m, which represents the efficiency of the motor compared with the NACC rating as unity, and by e_t which represents the efficiency of the transmission system, so that the formula becomes

$$\text{(B)} \qquad VC = \frac{8.4nb^2sR}{DW} \times e_m \times e_t.$$

VC is the ratio of the effort in pounds which the motor can exert at the driving tires to the total weight in pounds to be moved. If multiplied by the weight W, it will give the actual propelling force in pounds exerted at the tires of the driving wheels. VC is the measure of the ability of the motor vehicle to propel itself. It represents the very important inter-relation of the motor displacement, the gear reduction, the diameter of the driving wheels and the total weight to be moved. The motor speed at any given vehicle speed is determined by the relation of R and D. This formula further affords a ready means of comparing the practical effectiveness or ability of motor vehicles to perform their transportation function.

The terms e_m and e_t may not be definitely known, but can be quite accurately determined experimentally and for the average well-built car the product of these two terms will be about 95 per cent. If this factor be used formula (B) becomes

$$\text{(C)} \qquad VC = \frac{8nb^2sR}{DW}$$

The author gives the results of applying (C) to a number of well-known cars using W as the weight of the car with full road equipment plus 150 pounds for each person the car is designed to carry. The above formulae are intended to be used primarily for the reduction on high gear, but can be applied to lower gear or intermediate reductions after making allowances for an adidtional loss in the gear box. The author also considers in detail the various resistance items such as road surface resistance, grade resistance and wind resistance. (7 pp., *gpt.*)

SOUTH AFRICAN INSTITUTION OF ENGINEERS

Vol. 13, no. 11, June 1915, Johannesburg, South Africa.

Test of the Largest Air Compressor, G. M. Clark
The Influence Moisture in the Air has on Mine Ventilation, Arthur C. Whittome (abstracted)
The Centrifugal Pumping Plant of the Durban Roodepoort Deep, Limited, E. G. Izod and A. P. Rouillard

The Influence Moisture in the Air Has on Mine Ventilation, Arthur C. Whittome

Discussion of the amount of air which has to be supplied per man, the influence of the presence of water vapor on that man and the effects on mine ventilation. As an example, a case is taken for which certain assumptions are made, and it is shown how the amount of moisture affects the results obtained.

The objects of ventilation (in particular, mine ventilation) are, *first*, to sweep all deleterious gases out of the ventilated zone; *second*, to supply to the working places such weight of oxygen as may be required by the workers therein, and *third*, in some cases, to lower the temperature of the working places to such a degree as to make work possible in spots which would otherwise be too warm. The author uses the term "oxygen stuff" to denote the pure dry-air portion, consisting of oxygen and nitrogen, of either dry or saturated air. He compares data on the amount of air breathed in by a grown person hard at work as given by various authorities, but points out that the primary object of ventilation is the provision at every place of such a weight of oxygen stuff as will insure an atmosphere suitable for breathing and not merely a supply of a stated volume of air.

This makes the question of the presence of moisture in the air of high importance, since as the author has shown in tables and curves, under a given barometric pressure, oxygen stuff decreases and water vapor increases in weight per cubic foot as the temperature rises. Further, as the temperature of the saturated air rises, the elastic force of the

water vapor increases, and, therefore, with saturated air at a constant pressure, the elastic force of the oxygen stuff will gradually decrease with increasing temperature, showing the second cause for the decreased weight of oxygen stuff in saturated air. The weight of saturated air is lower per cubic foot than that of dry air at the same temperature and pressure, and with constant pressure and increasing temperature, saturated air decreases in density at a more rapid rate than does dry air. To illustrate the effects of moisture on mine ventilation, the author takes a case for which he makes the following assumptions: *a*. The collars of the upcast and downcast shaft are located on the same horizon, the shafts being sunk to a vertical depth of 3000 ft. and connected by a single drive at the 3000 ft. level; *b*. In each shaft, the mean barometric pressure is 13 lb.; *c*. Mean temperature in the upcast shaft is 90 deg. fahr., and in the downcast shaft, 60 deg. fahr.; *d*. The following alternative contents of humidity are assumed as being possible: *first*, the air remains dry during the passage through the mine workings; *second*, the air is kept constantly 100 per cent saturated at all points in its passage, and *third*, the air is dry during its passage through the downcast and saturated during its passage through the upcast shaft.

An investigation of the conditions prevailing in such a case shows that in the event of a constant mean temperature in the downcast shaft and a second constant mean temperature in the upcast shaft, the following rules will hold good: *first*, the least weight of oxygen stuff flows when the air throughout both shafts is perfectly dry; *second*, the greatest weight of oxygen stuff flows when the air in the downcast shaft is perfectly dry, and that in the upcast shaft is 100 per cent saturated; *third*, increasing the degree of humidity of air in the downcast shaft without lowering its temperature decreases the weight of oxygen stuff flowing; *fourth*, increasing the degree of humidity of the air in the upcast shaft without lowering its temperature increases the weight of oxygen stuff flowing; *fifth*, the greatest intensity of draft produced by natural means is obtained when no heat is abstracted from the air itself except in the downcast shaft (i.e., when water is only sprayed into the air into the downcast shaft, which does not mean, however, that the author advocates the disuse of sprays and water blast where these are necessary for the laying of dust or the absorption of deleterious gases).

There is one more aspect to the question of the presence of moisture in the air. It would seem that when a person breathes saturated air at a temperature above blood heat, there must be condensation of water vapor in his lungs and the resultant water, (almost certainly carrying deleterious gases and matter if the person is in a mine) must have a detrimental effect on the lungs and retard them in the efficient performance of their function. The author thinks that what is generally attributed to "laziness" developed by dark places and the impossibility of continuous supervision, should really be ascribed to the inability of the lungs to handle the extra volume of air which is requisite to provide the necessary weight of oxygen, the lungs being to some extent waterlogged by condensed water vapor and by the moisture which would normally be absorbed by non-saturated air in its circuit in the body, but which cannot be taken up by the saturated air.

The following are some of the further conclusions arrived at by the author:

1. Air should be cooled as much as possible before it goes to the downcast shaft; while it is being cooled, it should be kept as dry as possible unless the heat used to evaporate the water is abstracted from the air, thus lowering its temperature and making it denser.
2. Water having been once used for spraying purposes in situations where deleterious gases are absorbed should not be used again for spraying, as some of the entrained gases will be released to the atmosphere when a portion of the water is evaporated.
3. Carbon dioxide absorbed by the spray water should be removed therefrom before the water goes to the pumps, in order to avoid corrosion in the pumps and piping.
4. If there are ill effects caused by inhaling hot saturated air, the air should be cooled before going to the working places. This could be effected by the installation of small electrically driven refrigerating plants at necessary points in the mine, and passing the ventilating air through cooling chambers. (8 pp., 4 sheets of tables and charts. *peA*.)

CLASSIFICATION OF ARTICLES

Articles appearing in the Survey are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general; *h* historical; *m* mathematical; *p* practical; *s* statistical; *t* theoretical. Articles of especial merit are rated *A* by the reviewer. Opinions expressed are those of the reviewer, not of the Society. The Editor will be pleased to receive inquiries for further information in connection with articles reported in the Survey.

NECROLOGY

JOHN BROWN HERRESHOFF

John Brown Herreshoff was born in Bristol, R. I., in 1841. An attack of infantile glaucoma destroyed his sight, but his education was carried on in the schools of his native town. His bent for mechanism revealed itself at an early age, and the handicap of total blindness seemed to serve as a spur to his tireless energy. The miniature craft that he built and rebuilt in those early years served as the foundation of the skill and knowledge that he later turned to such good account.

At the age of twenty, he began the construction of larger craft, and in 1863 he embarked on naval construction as a business, which he carried on for more than fifty years.

In 1872, he entered into partnership with his brother, and together they designed and constructed for the U. S. Government the first torpedo boat, The Lightning.

In 1879, Mr. Herreshoff and his brother incorporated the Herreshoff Manufacturing Company, of which he was president and treasurer from its formation to the time of his death.

In 1892, the Herreshoff Manufacturing Company took up the designing and construction of yachts, with special reference to building craft that would enable the New York Yacht Club to hold the trophy against attack. The record of the Vigilant, Defender, Columbia, Reliance and Resolute indicate to what height of perfection Mr. Herreshoff's firm has carried the science of naval designing and construction.

Mr. Herreshoff was a member of the Institute of Naval Architects of London and a member of the Society since 1884. He died on July 20, 1915.

WILLIAM H. GERRISH

William H. Gerrish was born in Lowell, Mass., in 1865, and was educated at the Lowell High School and the Massachusetts Institute of Technology. He served his apprenticeship at the Lancaster Slate Company, Lancaster, Mass., from 1888-90. For several years he held a position in the drawing room of the Massachusetts Cotton Mills, Lawrence, Mass., and at the Fulton Bag Mills, Atlanta, Ga.

During the Spanish-American War he was associated with the ordnance department of the War Office, Washington, D. C.

In 1899, he became superintendent of Barber Flax Spinning Company, Paterson, N. J., from which he went to the Dolphin Jute Mills, Paterson, N. J. Four years later, he came to New York as superintendent of Commercial Twine Company.

In 1910, he was appointed smoke inspector for the State of Massachusetts, which office he retained to the date of his death.

He was a member of the Order of Masons and the Royal Arcanum. He died on July 15, 1915, in Malden, Mass.

EDWARD THOMAS MORRIS

Edward Thomas Morris was born in Lutterworth, England, in 1863. He was educated in public and private schools in England, after which he served his apprenticeship at Ruston & Proctor, engineers and boiler makers, Lincoln, England, from 1880-1884. During his apprenticeship he took night technical courses.

From 1884 to 1885, he was under instruction at the firm of J. & J. Thompson, marine engineers and boiler makers, Glasgow, Scotland.

From 1887 to 1888, he was outside erector of machinery for the Union Iron Works. From that date until 1899 he was superintendent of the marine engineers department of the Union Iron Works, having charge of the installation of boilers and machinery on the U. S. S. Charleston, San Francisco, Monterey, Olympia and Oregon.

From 1899 to 1900, he was assistant to the marine superintendent of the Pacific Mail Steamship Company, and from 1900-1902, general superintendent of the Fulton Engineering and Shipbuilding Works.

During the years 1902-1905, he was superintending engineer of the Oceanic Steamship Company, and in 1906 he became superintendent of construction with the Tracy Engineering Company.

From 1908-1911, he was engineer of construction for the Associated Pipe Line Company. In 1911, he was appointed manager of the Pipe Line Department of the Associated Oil Company, in which position he was in charge of the construction and operation of pipe lines and pumping stations of both companies.

Mr. Morris died in Oakland, California, on May 14, 1915.

GEORGE RIPLEY STETSON

George Ripley Stetson was born in Brooklyn, Conn., in 1837, and was the son of parents who were prominently identified with the anti-slavery and other reforms of the time. His early education was received in Brooklyn, Conn., and in Florence, Mass. His subsequent education was gained entirely through his own efforts without attendance at school.

In 1855, he entered a machine shop in Northampton as an apprentice. After completing his apprenticeship, he engaged in the manufacture of britannia ware. In 1864, he entered the employ of O. F. Winchester, manufacturer of rifles and ammunition, where he remained eight years.

In 1877, Mr. Stetson became superintendent of the Morse Twist Drill and Machine Company, New Bedford, Mass.

While in the service of this company and the O. F. Winchester Company he patented a number of devices which proved of value, receiving thirty-six patents for his inventions.

In 1890, Mr. Stetson became president and general manager of the New Bedford Gas & Edison Light Company.

He was a member of the Sutton Commandery, K. P., and a charter member and past patron of the New Bedford Chapter, No. 49, O. E. S. He was an active member of the National Electric Light Association and the Association of Edison Illuminating Companies, frequently representing these organizations as a delegate, and a charter member of the Society. He died on July 26, 1915.

HAROLD BENTLEY ANDERSON

Harold Bentley Anderson was born in 1878. He received his technical training at the Case School of Applied Science in Cleveland, Ohio, and while in college worked up a motorcycle and also a sewing machine driven by a small gasoline engine. By the time he left college he had completed an automobile and in 1901, he secured patents on two automobiles. He at once entered the employ of the American

Bicycle Company in Toledo, producing a system of double acting brakes and the steering gear that was used on the copy of the "Lifu" truck, made in Toledo.

In 1902, he became associated with The Winton Company as personal engineer for Mr. Winton. In 1904, he was made chief engineer of The Winton Company, which he held at the time of his death.

Mr. Anderson was a member of the Society of Automobile Engineers, American Institute of Metals, American Society for Testing Materials, International Association for Testing Materials, Cleveland Engineering Society, Beta Theta Pi, Theta Nu Epsilon, Cleveland Yacht Club, Cleveland Automobile Club and The Aeronautical Society. He was also an officer of the aero squadron of the First Aviation corps, New York City. He died on July 13, 1915.

PERSONALS

Edwin G. Hatch, consulting engineer, New York, will have charge of the inspection and testing of a 9000 k. v. a. transformer now building at the Westinghouse Electric & Mfg. Company's plant, East Pittsburgh, Pa., for the Victoria Falls and Transvaal Power Company of London. Five similar transformers were shipped to the company's plant in South Africa in 1912.

Dr. Addams Stratton McAllister resigned on August 1 his position as editor of the "Electrical World," New York.

F. M. Feiker, formerly associated with "Factory" and "System" and for the past few weeks with the "Electrical World" editorial staff, will succeed Dr. McAllister as editor of the "Electrical World."

Morris Knowles, consulting engineer, has acquired the engineering business formerly conducted from offices in Pittsburgh, Pa., and Canton, Ohio, by Mr. L. E. Chapin, recently deceased. Mr. Knowles, who was previously associated with Mr. Chapin, will conduct the combined business from his office 2541 Oliver Building, Pittsburgh, continuing to specialize in water works, water power, sewerage, sewage treatment and disposal sanitary investigation, town planning, flood prevention and valuation work.

Joseph A. Whitlow has been appointed chief of the Department of Gas, Electricity, Heat and Water for the Public Service Commission of Missouri. He was formerly chief engineer of the Board of Education of St. Louis.

H. T. Woolson, formerly chief engineer of the Gas Engine & Power Company and the C. L. Seabury & Company Consolidated has accepted a position as truck engineer of the Packard Motor Car Company, Detroit, Michigan.

Frank B. Gilbreth who is conducting a three weeks course in motion study in Providence, R. I., recently held an annual conference with a number of educational men and women from all parts of the country who are interested in the problems of scientific management. The meetings of this year have special significance in that Mr. Gilbreth has been asked by the governments of England, France and Germany to study the problems of efficiency with the view to considering the opportunities which are open to the mutilated soldiers in the present war.

Dr. John A. Brashear, President of the Society, has been chosen by Gov. Brumbaugh as the greatest Pennsylvanian. This selection was the result of an invitation from the officials of the Panama-Pacific Exposition to select a man upon whom the exposition could confer an honor as being eminent in the life of Pennsylvania. The governor not wishing to make the selection himself, decided to let the choice rest with the editors of the state and sent a letter to the journalists asking them to make the selection, and as a result Dr. Brashear was chosen.

EMPLOYMENT BULLETIN

The Secretary considers it a special obligation and pleasant duty to be the medium of assisting members to secure positions, and is pleased to receive requests both for positions and for men. Copy for the Bulletin must be in hand before the 18th of the month.

POSITIONS AVAILABLE

The Society acts only as a "clearing house" in these matters and is not responsible where firms do not answer. In sending applications stamps should be enclosed for forwarding.

0205 Draftsman experienced in the designing of bending rolls. One who has actually done work of this character. Location Southern States.

0206 Young engineer for research work of experimental nature, along general methods which have been partially developed. Must have fairly good command of mathematics and particularly adapted to contrive and carry out the mechanical work incident to the experiments. Position offers exceptional opportunities for the development of individual ability. Approximate salary to start will be $75.00 per month. Location Philadelphia.

0208 Designing engineer in drafting room of Boston concern. Prefer young man with some experience on conveyor work, though such experience is not absolutely necessary.

0209 Salesman for New York concern manufacturing rubber belting and rubber goods for mechanical and manufacturing purposes.

0210 Designer of special and automatic machinery and production tools. Give age, experience, and salary expected. Location New York.

0211 Young engineer with two or three years experience, for work in superintendence and operation of buildings. Location New York.

0213 Two designers with initiative in development work, machinery design, etc. Familiarity with automobile work desirable but not essential.

0220 A capable man, preferably with engineering education and good shop training capable of specifying equipment, to sell metal working machine tools in the Middle West.

0221 A man familiar with presses and forging machinery capable of selling same. Location Middle West.

0223 Department of Mechanical Engineering, Canadian University, invites applications for the position of lecturer in mechanical subjects, mechanics of machines and machine design, or shop practice and laboratory assistant. Applications should be accompanied by a statement of age, qualifications and salary expected.

0225 Mechanical draftsman; some experience in cane or beet sugar refinery; for two or three months work. Salary $5.00 per day. Location Louisiana.

0226 For firm in New York State, experienced mechanical engineer, who is thoroughly posted in stamping and drawing propositions.

0227 Designer with a broad experience on metal-working machinery and tools wishes to get in touch with a concern which would undertake the manufacture of new line of medium sized machines. Location Middle West.

0228 Draftsman, American; familiar with design and shop work. Position part time in New York State and later in New York City. In applying, state experience, etc. Salary $100.00.

0229 Inspector of machinery same line as 0228. Apply same as position 0228.

0230 Factory manager of more than ordinary experience in automobile manufacture for a company organizing for manufacture of automobiles. Name confidential.

0233 The direction of fire and patrol service about to be reorganized along most modern lines by a large automobile manufacturing company, is now open to a man who is qualified to take charge, who can shoulder the responsibility of custodian of their huge investment in plants, machinery and stock, with absolute integrity and fearlessness; a sane and healthy view of human nature; alertness rather than suspicion; understanding of various nationalities and types of men and their ways; ability to select a body of good men; to maintain proper discipline and to communicate a large sense of responsibility. Address 0233, care of The Am. Soc. M.E.

0234 A large industrial corporation manufacturing a diversified product including automobiles, desires a works manager for its machine shops, foundries, forge plants and other factory interests. A man of proven ability as an operating executive is sought; his record must demonstrate that his abilities combine a wide mechanical experience with the gift of organizing and obtaining the best from large bodies of men, a selective insight into efficiency methods, an analytic grasp in handling a large mass of detail, a true conception of how and what to delegate and a comprehensive understanding of the problems of the employee. Address in strictest confidence, P. O. Box 450, New York.

0235 Master Mechanic for a cotton piece goods finishing plant. First and principal requirement that he be well educated and preferably a technical graduate. Experience in finishing business not necessary, but ability to handle men and lay out the work of a machine and carpenter shop, piping and electrical department essential. Excellent opening for a man of ambition and energy. Location New England. Apply by number.

0236 Plant Manager for large silk weaving mill, who can lay out buildings and power plant and take charge of maintenance. Apply by number.

0237 Young graduate mechanical engineer to go into heat, light and power department of a plant employing about 1200; will be expected to be able quickly to take charge of the both steam and gas engine departments. Apply by number, giving age, education, experience and salary expected. Location New England.

0238 Young graduate engineer preferably with some experience in machine shop or construction work, to go into the time study and rate fixing department of a large plant in New England; will be expected to learn the business and will then be promoted into the operating departments in an executive office. Apply by letter, giving age, education, experience and salary.

0239 Young man, preferably one who has attended an engineering college, with a knowledge of machine shop practice, to enter the repair department of a large concern as assistant to the master mechanic. Location New England.

0240 Wanted, superintendent for munition shop, one with sufficient experience in the manufacture of six inch (and over) high explosive shells, to design a building, equip it with suitable machinery and operate the same after completion. Plant to be built within 25 miles of New York. Give references as to ability and experience. All communications confidential.

MEN AVAILABLE

The published notices of "men available" are made up from members of the Society. Notices are not repeated in consecutive issues of the Bulletin. Names and records are kept on the office list three months, and at the end of such period if desired must be renewed.

I-240 Member, who has designed, built, installed and operated hydraulic, conveying, woodworking, hoisting, computing, combustion, power transmission and other apparatus, mechanical and electrical, wants position with thoroughly established corporation in N. E. or near N. Y. City, in which a good salary may be earned several times over by the systematic introduction of economies and "short cuts" in men, methods and machinery between coal pile (or water power) and shipping room. Immediate salary of less importance than congenial position in which brains, experience and loyalty will assure fair returns to self and family based on results obtained.

I-241 Works manager or general superintendent, mechanical engineer, graduate of the Mass. Inst. of Technology, with 16 years experience in the manufacture of boilers. At present employed, having charge both of office and shop. Has produced results and can show a clean record. Address through the Society.

I-242 A mechanical engineer, technical graduate, with broad shop and mill experience, desires to become associated with an important New England concern. Full particulars exchanged.

I-243 Technical graduate, with seven years experience in design and construction, desires position with architect or engineer to take charge of service work including power plant, heating, plumbing, lighting and allied branches. Experience in railroad terminals and shops, theater, factory, church and residence work.

I-244 Technical graduate M. E., thirty years old. One year mechanical engineer with railroad. Six years experience with car manufacturer, designing dies, tools, heavy machinery, hydraulic presses; also projectile forgings. Good shop experience. Would be useful as assistant master mechanic or mechanical engineer. Location immaterial.

I-245 Member, age 37, sales engineer technical training, practical mechanic with a successful sales record; speaks Spanish and is conversant with South American trade conditions, forceful Spanish advertising, cataloguing, open for engagement in South American field with a responsible company manufacturing machinery, steel products, hardware or agricultural implements. At present connected with prominent Eastern jobbing house as department sales manager. Salary expected secondary to future prospects.

I-246 Position wanted as foundry superintendent or manager of works doing mostly foundry work; can furnish references satisfactory as to ability from previous employers. Practical foundry man who has filled all positions from laborer to the management of some of the most up-to-date companies in this country. Understands the business end as well as the practical end of the iron and brass foundry business.

I-247 Junior, technical education, with twelve years experience in designing, estimating, constructing and selling all kinds of mining, crushing, cement making machinery and complete plants, desires position.

I-248 Mechanical engineer, college graduate, associate member, 7 years practical experience, 31 years of age, married. Experienced in time study, rate fixing, steam engineer, master mechanic, mechanical superintendent. Salary $2,400.

I-249 Member, 28 years practical and executive engineering experience in designing, manufacturing, production, organization of manufacture, equipment, etc., modern shop practice and management, desires position as general superintendent or works manager, two or five year contracts only considered. Large experience in precision and quantity manufacture and expert in economical production.

I-250 Member of the Society, with eight years practical experience in general mechanical engineering, and four years experience in the installation of scientific management, wishes a position as efficiency engineer with a progressive concern.

I-251 Junior, age 21, two years at Cornell, two and a half years shop and drafting room experience, wants position in testing, engineering or manufacturing department where there is a good chance for advancement. Willing to work hard and study to obtain advancement. Salary to start, a secondary matter. At present employed.

I-252 Member, graduate Stevens Institute Technology, 13 years experience in design and construction, steam-electric plants, pumping stations, industrial works, heavy foundations, involving pneumatic caisson methods, reports and appraisals, also business experience, is open for engagement.

I-253 Sales engineer with seven years experience in power plant and water works fields; can furnish proofs of ability; would consider a similar position or connection with consulting engineer's office. New York City preferred.

I-254 Associate Member, technical graduate, age 33, nine years shop, erecting, designing and operating experience in the steam engineering field, having specialized in the design and manufacture of steam turbines; capable of taking full charge of same. Location immaterial. At present employed.

I-255 Junior Member, graduate mechanical engineer Stevens Institute of Technology; three years experience and now occupied in fire protection work, engineering and sales; familiar with insurance and factory conditions, desires to associate with insurance company, firm of brokers or insurance engineers having department making a specialty of improved risks. Location in New England preferred.

I-256 Junior member, technical school graduate, with five years engineering experience, 3½ years in present position as assistant mechanical engineer in large government engineering testing plant, desires position with testing department of large corporation, power station or engineering firm; capable of conducting complete tests of steam boilers, engines, turbines and all auxiliary machinery. Would accept $1800 to start.

I-257 Member, technical graduate, 12 years experience in design and supervision of heat, light and power plants, fuel and combustion, desires position as supervisor of power plants for efficiency, or designer with private corporation or consulting engineer. At present employed.

I-258 Graduate, mechanical engineer, associate-member, 4 years experience in designing, heating and ventilating plants; past 4 years in tests and experimental research work on steam boilers, gas, oil and steam engines, compressors, etc., as assistant professor in mechanical laboratory; at present employed in experimental department of oil engine maker; knowledge of German and French; desires to improve present conditions. Location immaterial.

I-259 Member, Cornell graduate, electrical and mechanical engineer; extensive experience in designing, constructing, operating, maintaining, inspecting, testing, and office engineering; desires position with engineering or manufacturing concern. Employed at present.

I-260 Member, 38, technical graduate, practical experience in various shop departments, also training from draftsman up to executive as treasurer and general manager; particularly well qualified for high-grade sales engineering; solicits opportunities from those seeking a country-wide market for product sold on quality rather than price basis. Has recently tripled volume of sales per dollar expended in repeated instances and has been particularly successful in the comprehensive organization and development of reliable sales forces. Would consider position of general manager if responsibilities were chiefly those of securing satisfactory market. Location immaterial.

I-261 Mechanical engineer, varied experience in technical advertising and editorial work, desires position as advertising manager of manufacturing concern. Location immaterial. At present employed.

I-262 Associate member, graduate of Massachusetts Institute of Technology, five years experience in laboratory and factory, wishes position as assistant to manager, superintendent or production engineer. Now employed. Opportunity first, salary secondary.

I-263 Member with broad manufacturing experience from apprentice to general manager in foundry, pattern, machine, blacksmith, boiler shops and drawing office, positions which called for initiative executive ability, tact, the handling of correspondence, organization of men and the direction of their work, wishes employment as manager, superintendent or other executive position. Location immaterial.

I-264 Associate-Member, graduate M.E., age 31, married, possessing strong personality with plenty of initiative and executive ability, having had seven years experience in machine shops, foundries, engineering, selling and efficiency work, desires to represent manufacturer in Chicago in a line of a technical nature where hard work and ability will produce a big volume of business. Available Oct. 1st.

I-265 Permanent responsible position wanted by an energetic, thoroughly practical and experienced shop and office engineer; Cornell M.E.; married; successful extended experience in the development of inventions, special and standard machinery, in plant management and as economy engineer.

I-266 Do you need the services of a man with a wide experience in the iron and steel industry? One who has had a good practical knowledge of shop and steel construction, as designer, engineer and contractor, for large independent steel companies, supplemented by three years as inspection engineer in charge of mill, shop and field inspection of structural steel, railroad rails and galvanized product. College graduate; age forty.

I-267 Member just returned from abroad would like to correspond with a well established concern (Middle West preferred) in view of securing a position as tool engineer or chief draftsman in full charge of the designing, building and manufacturing of tools, dies and machinery. Seventeen years experience with four of the largest and best known concerns in U. S. and Europe, on small, accurate interchangeable manufacture. Age 37, natural mechanical and executive ability, full references.

I-268 Junior, graduate, at present employed, five years design, manufacturing and power experience. Wants a live wire position.

ACCESSIONS TO THE LIBRARY

This list includes only accessions to the library of this Society. Lists of accessions to the libraries of the A.I.E.E. and A.I.M.E. can be secured on request from Calvin W. Rice, Secretary of the Am. Soc M. E.

ALUMINUM ELECTRICAL CONDUCTORS. *Pittsburgh.* Gift of Aluminum Company of America.

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A series of essays based on extensive travels in this country. The author devotes much space to a discussion of economic conditions as he saw them. W. P. C.

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THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

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[1] A complete list of the officers and committees of the Society will be found in the Year Book for 1915, and in the January and July 1915 issues of The Journal